THE DESIGN OF A 5 kW MICROHYDRO GENERATING SET

A thesis

submitted in fulfilment

of the requirements for the degree

of

Master of Engineering (Mechanical)

in the

University of Canterbury

by

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February 1992

In Memory of My Parents

ABSTRACT

In rural areas of many countries, the supply of electricity has always been scarce due to the relative isolation from the national electricity distribution system. In New Zealand for example, many high country farm houses rely almost entirely on diesel generating sets for power. This method of power generation costs more than urban electricity supply. However, a recent advance in control technology, and a new turbine concept have shown that very small hydro power schemes also known as microhydro, can now produce urban quality electricity at a cost not much greater than urban supply.

Through research and testing a 5 kW microhydro generating set was developed. The design of the microhydro generating set involves the utilisation of the latest control technology, and proprietary components. The single most important contribution to the development of the microhydro generating set was the use of an electronic load governor with the run-of-the-river principle which results in constant flow operation of the turbine. This eliminates the need for a precision speed governor and accurate turbine flow control and thus greatly simplifies the turbine design. A malfunction protection system and a load management system have also been incorporated to allow maximum utilisation of the power output without risk of damage to the generating equipment.

The use of a Delphi electronic load governor in conjunction with a brushless, self excited and self regulated synchronous generator have produced a stable electrical output of 5 kW at 230 V, 50 Hz, AC, when using an appropriate flywheel to provide shaft inertia to dampen out electrical transients. It was found that shaft inertia of four times that recommended by the governor manufacturer, was required.

One of the simplest turbines is a centrifugal pump operating in the reverse mode. Tests have shown that centrifugal pumps perform exceptionally well as power recovery turbines, with efficiencies similar or in some cases even greater than that for the pumping case without any modification. Consequently, the combination of well established mechanical and control hardware, simplifies the design and installation of a microhydro scheme to a technical task rather than one requiring the expertise of a design engineer.

A set of five machines was designed, covering the range of heads and flows most commonly encountered in hilly isolated areas. The operating conditions cover heads ranging from 30 m to 150 m and flows ranging form 30 l/s to 10 l/s respectively.

ACKNOWLEDGEMENT

I wish to express my sincere gratitude to my two supervisors Dr J.K. Raine and Mr E.P. Giddens as well as Mr R.T.C. Harman during the absence of the former overseas, who have all devoted their valuable time and effort to assist me with this project.

My appreciation is given also to the many technicians at the University of Canterbury who assisted me with my research on the practical side, and in particular, Messrs I. Shepperd, A. Poynter, K. Brown, O. Bolt, P. Smith and E. Harris, and also Mr J. Fahey (of B.R. Homersham Ltd, Christchurch) for providing us with the requisite equipment.

Thanks are due also to all my colleagues at Works Consultancy Services, Wellington especially Messrs O.R.A. Burgess, W.D.S. Brander and J.E.R. Fulton for their help. In addition I wish to acknowledge the encouragement I received from Mr R.H. Chilton for his tolerance in allowing me the resources and opportunity to complete the final phase of this thesis.

This acknowledgment would not be complete without reference to both financial support I received from Works Consultancy Services and , in the reassurances I received from the many friends I made whilst studying at the University.

Last, but certainly not least, I am indebted to the members of my family who sustained me during those years.

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LIST OF SYMBOLS AND ABBREVIATIONS

AVR	Automatic voltage regulator
B_1, B_2	Shear forces at turbine bearings (N)
BM	Bending moments (Nm)
ELG	Electronic load governor
F_u, F_y, F_e'	Ultimate, yield and endurance stresses (MPa)
F'	Braking force (N)
F	Reaction force due to braking (N)
f	Electrical frequency (Hz)
H_t, H_p	Turbine and pump head (m)
H_f	Friction head (m)
ΔH	Head loss (m)
I	Moment of inertia (kg/m^2)
J	Second moment of area
K_a, K_c, K_e	Shaft services and geometry factors
K_t, K_m	Shock factors for torsion
K_d	Draft tube friction factor
L_n	Bearing life (Hours)
L_e	Pipe equivalent length (m)
N	Rotational speed (RPM)
N_c	Critical speed (RPM)
N_{sp}, N_{st}	Pump and turbine specific speeds
n_s	Synchronous speed
NPSHR	Net positive suction head required (m)
NPSHA	Net positive suction head available (m)
P	Power (kW)
δp	increased pressure (kPa) or (m)
pf	Power factor
Q	Discharge (l/s), or (m^3/s)
R	Relative velocity between fluid and impeller (m/s)
Re	Reynolds number
r	Radius of impeller or runner (m)
T	Torque (Nm)
TREH	Total required exhaust head (m)
TAEH	Total available exhaust head (m)
u	Peripheral velocity of impeller (m/s)

v	Flow velocity (m/s)
v_w	Velocity of whirl (m/s)
ΔZ, Δz	Suction head (m)
τ_a, τ_m	Alternating and mean torsional stresses (MPa)
σ_a, σ_m	Alternating and mean bending stresses (MPa)
σ_r,σ_t	Radial and tangential stress components in flywheel (MPa)
ω	Rotational speed (rad/s)
μ	Friction coefficient
σ'	Von Mises stress (MPa)
δ	Shaft deflection (mm)
ξ	Pipework head loss coefficient
η	Efficiency (%)
ρ	Density of water (kg/m^3)
σ_c	cavitation constant

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CHAPTER ONE

1 INTRODUCTION

1.1 HISTORIC NOTES

Power from water is not a new technology, as it has been around since long before the discovery of electricity. In fact, it is the origin of the present conventional large scale hydro power systems. In the late nineteenth and early twentieth centuries many small scale hydro power schemes were developed, providing power for established urban areas. As the demand for electricity grew, the size of the power systems grew accordingly. Consequently, electricity became cheaper and more widely available and this made small scale hydropower schemes less attractive.

In recent years, there has been a growing realisation of the importance of electricity in bringing social and economic development to rural communities in the less developed countries of the third world, and in the rural areas of the developed countries with relative isolation from the national distribution system. The need for standard urban quality electricity supply at not much greater cost reverts the idea back to the old approach of using very small scale hydro power schemes. In many countries, there are extensive areas, with plenty of small scale hydro potential in need of development [28,29,43].

1.2 DESIGN CONCEPT

For very small scale hydro systems to become economically viable, they must not be viewed as a scaled down version of the conventional hydro power scheme. A completely new concept must be adopted. It is envisaged that the new technology for very small scale hydropower plants should be based on the following concept:-

1. Avoidance of the need for professional expertise
2. Use of newly developed control technology
3. Use of proprietary components

The implication of avoiding the need for professional expertise is that very small scale hydro power plants can be brought within the means of local people to install, operate and maintain. The use of new developments in system controls makes for a simpler, cheaper but still reliable generating system. Finally, use of proprietary components implies ready availability and interchangeability of parts.

1.3 MICROHYDRO POWER SCHEMES

Modern conventional hydro power schemes of 1 to 2 MW are considered to be at the lower limit of size from an economic view point when considering a national electricity distribution system. Mini and micro hydro power are terms used to describe small and very small schemes. Hydro power schemes of less than 100 kW are usually considered to be within the range of microhydro power.

In this project, microhydro power is reserved for small, privately owned schemes because in terms of financing a change in emphasis is encountered between public ownership and private ownership. In privately owned schemes, capital cost and payback period are very important. The implication is that the purchase and operation of the scheme must come within the resources of the individual, i.e. a farmer. This is the philosophy which guides the whole design.

1.4 APPLICATION OF MICROHYDRO

Currently, the most common form of power in isolated localities is diesel-driven generators. However, there are many disadvantages associated with diesel generation, such as high operating costs, high maintenance costs, and the cost per unit of electricity escalates as the consumption decreases. At low load factors, diesel generators are inefficient and expensive, and consequently the practice is to use them only intermittently.

Microhydro power, on the other hand, incurs no fuel cost and is available 24 hours per day. Initial capital cost may be higher than that of diesel but the much lower operating cost makes microhydro power more attractive in the long term.

1.5 A TYPICAL MICROHYDRO INSTALLATION

A typical microhydro installation comprises principally an intake, a pipeline and generating equipment. The system usually operates on a 'run-of-the-river" principle, (ie with no storage), diverting the necessary flow through the pipeline to develop the head necessary to generate the electrical output. Within the generating equipment, a turbine converts the head to rotating shaft power which in turn drives a generator to produce the electrical output supplying the consumer reticulation.

One important contribution to the cost of microhydro generating systems is the inexpensive electronic load diverting governor which controls the frequency by controlling the power consumption. It diverts any unused electrical power to waste as opposed to the conventional governor which instead manages the input water power to the turbine for the same purpose. With the absence of the requirement to regulate the water power at the turbine, the simplest of turbines can be used. These include centrifugal pumps in the reverse mode of operation. The use of pumps not only removes the principle source of complexity in the governing of the turbine but opens the way to a system which allows proprietary components to be used successfully, thereby reducing the complexity of installation to the point where technical skills alone are sufficient.

Microhydro schemes in general have to cope with a variety of water resources and these can range from large heads with low flows to small heads with large flows.

The general arrangement of a medium head microhydro scheme is shown in Fig 1.1.

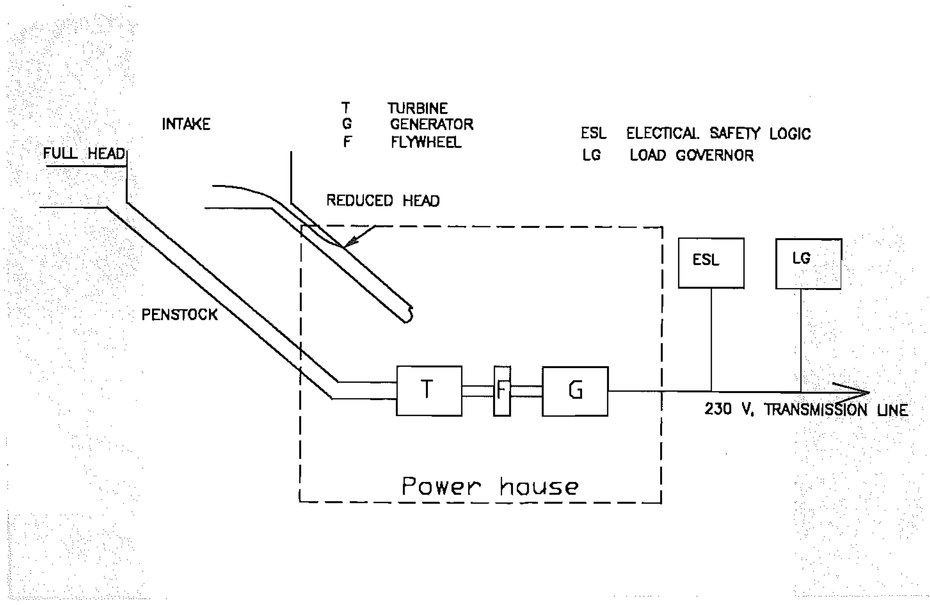


Fig 1.1 A typical medium head microhydro scheme

From the figure above the typical microhydro generating scheme using modern control technology would be as follows:-

1. Intake
2. Pipework
3. Turbine
4. Synchronous generator
5. Coupling/Flywheel and brake
6. Electronic load diverting governor
7. Electrical safety logic
8. Electrical distribution system

The isolated microhydro systems presently require the use of synchronous generators rather than induction generators [2]. With the synchronous generator, excitation and voltage regulation systems are built in, whereas with the induction generator, external excitation and voltage regulation have to be provided. The induction generator is normally used to feed into a network in which the frequency and voltage are established independently of the induction generator.

One other important aspect of any microhydro installation is safety. A malfunction protection system proposed by Giddens and Bryce [3] allows microhydro plants to operate unattended. The basic principle is to protect appliances connected to the system and the generator itself, from abnormal conditions.

1.6 THE PACKAGED 5 kW MICROHYDRO GENERATING SET

Based on the concepts outlined earlier, it is envisaged that the proposed microhydro generating set will take the form of Fig 1.2.

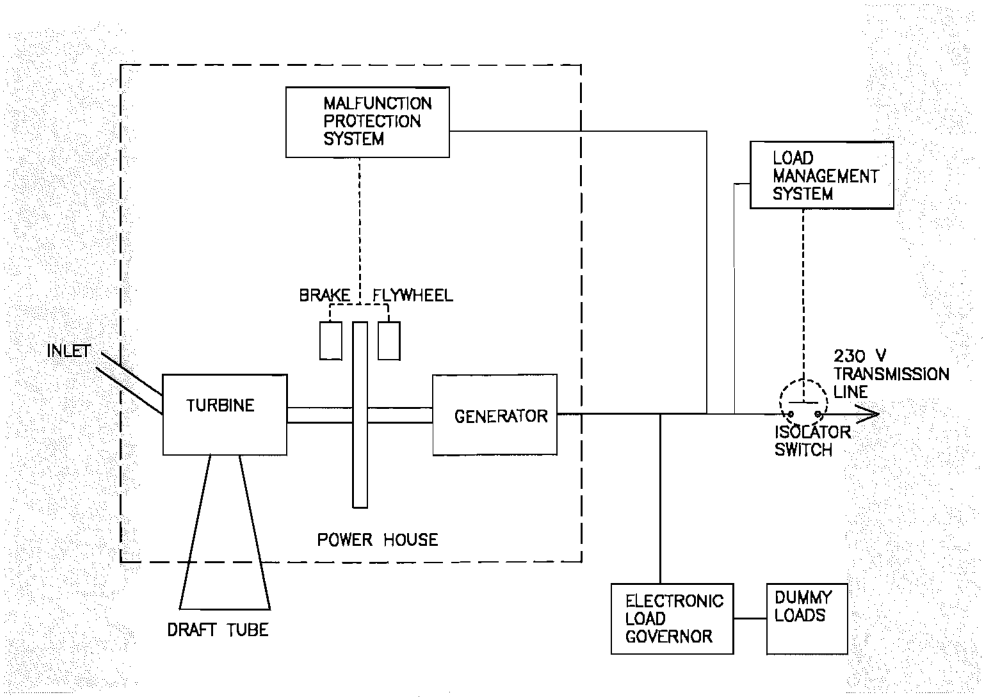


Fig 1.2 A packaged microhydro generating set

The packaged microhydro generating set comprises:-

1 Turbine (centrifugal pumps operated in reverse)
2 Generator (brushless synchronous generator)
3 Coupling/Flywheel and brake
4 Electronic load governor
5 Electrical malfunction protection system
6 Load management system
7 Inlet and outlet pipework including the draft tube

The principle of operating the packaged microhydro generating set is similar to that of a typical microhydro installation illustrated earlier. The turbine converts the head of the water to shaft power which in turn drives the generator to produce a single phase electricity supply. The draft tube is necessary to recuperate as much as possible of the remaining kinetic energy in the water. The speed, and therefore the frequency, is controlled by an electronic load governor which controls the total electrical demand presented to the synchronous generator, and automatically diverts the unused electrical load to banks of dummy resistive loads. The switching of the governor to control the frequency, in response to the changes in consumer load, is smoothed by the inertia of the flywheel.

In this particular design, the flywheel serves two purposes. One is to facilitate the governor control system, and the other is to accommodate the fail-safe braking system used to stall the machine in the event of a malfunction. The malfunction protection system protects the generator and consumer appliances connected to the system from extremes such as over/under voltage, over/under frequency and over current. A load management system is also provided which protects the consumer from nuisance tripping when trying to make full use of the power available. On overload, a warning is issued to the consumer by visual or audio means to shed load. If overload continues, supply will be disconnected, and all power is then diverted to the dummy loads.

It is envisaged that the proposed design will offer an attractive manufacturing proposition and will be easily operated and maintained by the owner. This is seen as being something involving maximum standardisation with sufficient flexibility. The flexibility is necessary for the different site conditions ranging from large head/low flow to low head/large flow. Standardisation is also important for limiting components and minimising manufacturing costs.

Electrical appliances designed for use on standardised power supply systems are available world-wide. The systems also supply power on a continuous 24 hour per day basis and this represents the quality supply that the microhydro power schemes need to produce. To satisfy the needs of New Zealand consumers, a standard electrical supply of 230 volts, 50 Hz, AC, should be available, with a good standard of reliability and at a price which is acceptable to the isolated consumer.

Studies on New Zealand urban residential electricity consumption have established that maximum demand in residential housing is approximately 12 kW, and the average 24 hour demand is approximately 1.5 kW. To design a generating system to satisfy maximum demand would be to too expensive, and to design for average demand would be impractical. It is therefore proposed for the purpose of this project, that a bench-mark figure of 5 kW be adopted for the example of our isolated New Zealand farm house [22].

1.7 SCOPE AND OBJECTIVES OF THE PROJECT

The objective of the present work is to establish the design for a packaged unit comprising the turbine, generator and controls but excluding the intake, pipeline and the electrical distribution system. Other studies have already established a suitable design for a water intake for pipelines and for matching of the electrical output to that of the customers' requirements [B5,B6].

The detailed objectives of the project are as follows :-

Chapters 1 and 2	To review the current research in the field of microhydro.
Chapter 3	To study the mechanics of fluids, with particular emphasis on Francis turbines and centrifugal pumps as turbines.
Chapter 4	To select and test appropriate centrifugal pumps to be used as turbines.
Chapter 5	To select appropriate generators to be used, and to assess the advantages and disadvantages of synchronous and induction generators.

Chapter 6	To study the function and operations of an electronic load governor, and to investigate its transient response characteristics in conjunction with a brushless self excited synchronous generator.
Chapter 7	To develop a malfunction protection system and a load management system.
Chapter 8	To design a packaged microhydro generating set to the required specification.

CHAPTER TWO

2 MICROHYDRO SYSTEMS - FUNDAMENTALS AND LITERATURE REVIEW

2.1 RESEARCH INTO MICROHYDRO TECHNOLOGY

Studies on microhydro in developing countries such as Papua New Guinea and India [29,33] have sited a number of drawbacks including:-

1. high capital cost
2. maintenance problems
3. disputes over ownership and sharing of benefits
4. unsuitability of electrical energy as a form of domestic energy supply
5. lack of industrial load to raise utilisation to a reasonable level
6. excessive dependence on scarce engineering skills

On the positive side, however, studies by Gafiye and Robinson [20,54] have shown that microhydro power can be a viable source of energy if properly designed. Thus, to make microhydro power a viable alternative both in developed and developing countries, the high capital costs, the most significant obstacle to their establishment, should be reduced to a more acceptable level. Technical and cultural difficulties may also need to be overcome, and these should all be addressed at the design stage of the microhydro system.

It is envisaged that microhydro power may become economically viable if the need for expertise is avoided. This may be achieved by the use of modern control technology and the use of proprietary components.

The scope of this chapter is to review the technology of microhydro in general with particular attention paid to the use of electronic load governors and the use of pumps as turbines, and various general aspects related to microhydro technology.

2.2 LOAD GOVERNING

One of the significant advances which has been made in recent years is the adoption of a radically different approach to speed control. Formerly, complex mechanical water flow governors were used to control the turbine speed and thereby the voltage and frequency of the electrical output. Their mechanical complexity and unreliability contributed greatly to the cost of microhydro projects.

However, electronic load governors such as that developed by Delphi Industries Ltd [11,59] are available and are acknowledged as the most reliable method of controlling microhydro plants. The electronic load governor, being almost maintenance free and easy to operate, has the added advantage of allowing the use of simpler turbines including centrifugal pumps in reverse, giving an overall reduction in the cost of the plant. Other electronic load-diverting governors are available from various sources around the world [5,52].

The principle function of the electronic load governor is to maintain constant frequency of the electrical output by diverting unused electrical load to waste. The Delphi governor adjusts the diverted load in 15 steps from no-load to full-load on the basis of a frequency range of 49.3 to 50.7 Hz. It can be used with loads up to 12 kW, but can be used in multiples for greater loads. The operation and testing of the governor is outlined in Chapter 6.

With a constant supply of power to the turbine, any variation in total electrical load will produce variation in current, voltage and frequency, and any one of these could be used by the governor as a reference signal to control the load. However, the most convenient signal is considered to be frequency. Since the frequency is uniform throughout the system, the electronic load governor can be located anywhere on the electrical reticulation.

2.3 TYPE OF TURBINES

a) Centrifugal pumps as turbines

Centrifugal pumps can operate effectively as turbines. Numerous papers have been written on the subject, but apart from one or two well established applications [4,55], limited use has been made of this knowledge. Characteristics of pumps are freely available from all pump manufacturers, but their characteristics as pump turbines are not. As explained by Stepanoff [57] any centrifugal pump can become a turbine. However, Engenda [15,16] stresses that the use of pumps as turbines is to be recommended for certain specific applications only:-

1 where initial cost is more important than efficiency and wider operating range,

2 where flow is more or less constant,

3 where tested turbine data for the pump to be used are available. (Although the data is not readily available, it may be obtainable by test once the application is clearly established),

In other cases, a conventional turbine of the same output would be a better choice.

b) Operation of pumps as turbines

In the normal operation of a pump, energy is added to the elements of the flow by action of the impeller. The performance of the pump is the result of integrating this energy increment for all fluid elements passing through the pump and taking into account the losses of energy and flow occurring between the inlet and outlet of the pump.

Owing to the non-uniformity of the flow through the impeller, all fluid elements will not receive the same increment in energy nor experience the same losses of energy. The head developed by the pump will be an integral of the net addition of energy to the flow.

In general, due to losses, the pumping head developed is less than the energy added by the impeller.

All elements of flow passing through the impeller will not pass straight through the pump due to recirculation through the impeller clearance spaces. There are also the problems of non-uniform velocity distributions, of flow separation, and even the possibility of pockets of reverse flow occurring within the impeller itself. The net flow out of the pump will be less than the total flow acted upon by the impeller.

In general, due to losses, the pump flow is less than the flow through the impeller

It is to be noted that in both respects (energy transfer and flow efficiency) small commercial pumps are less efficient than large, carefully designed and custom-made pumps.

Now, if losses were ignored and the process regarded as conservative it would be reversible by definition. This reversibility would require reversing both the flow direction and the direction of rotation of the impeller for a reversing of energy flux to occur. When the machine is actually operated in reverse mode to produce the reverse function, the flow through the impeller will be similar in principle but different in detail.

Losses of both energy and flow are inevitable and do occur. Thus, the turbining head is greater than the energy imparted to the impeller, and the turbining flow is greater than the flow through the impeller.

It follows that, for reverse operation at the same speed, and in both cases at the best efficiency point (BEP), the turbining head is greater than the pumping head and the turbining discharge is greater than the pumping discharge.

c) Pump turbine performance characteristics

A comparison of the characteristics of normal pump operation with the characteristics of the same pump operated as a turbine at the same speed is shown in Fig 2.1. The curves are normalized by the value of head, flow, efficiency and power at the pump BEP. Note that the location of the turbine BEP is at higher flow and head than the pump BEP. The ratio of the turbine capacity to the pump capacity at the BEP and the turbine head to the pump head at the BEP have been observed to vary with specific speed. Ratios of 1.1 to 2.2 have been determined by test [56]. Moreover, the turbine maximum efficiencies tend to occur over a wide range of capacity. In other words, the efficiency curve for a turbine is flattish near its maximum. Consequently, relatively wider ranges of turbine operating head can be accommodated without an adverse effect upon efficiency.

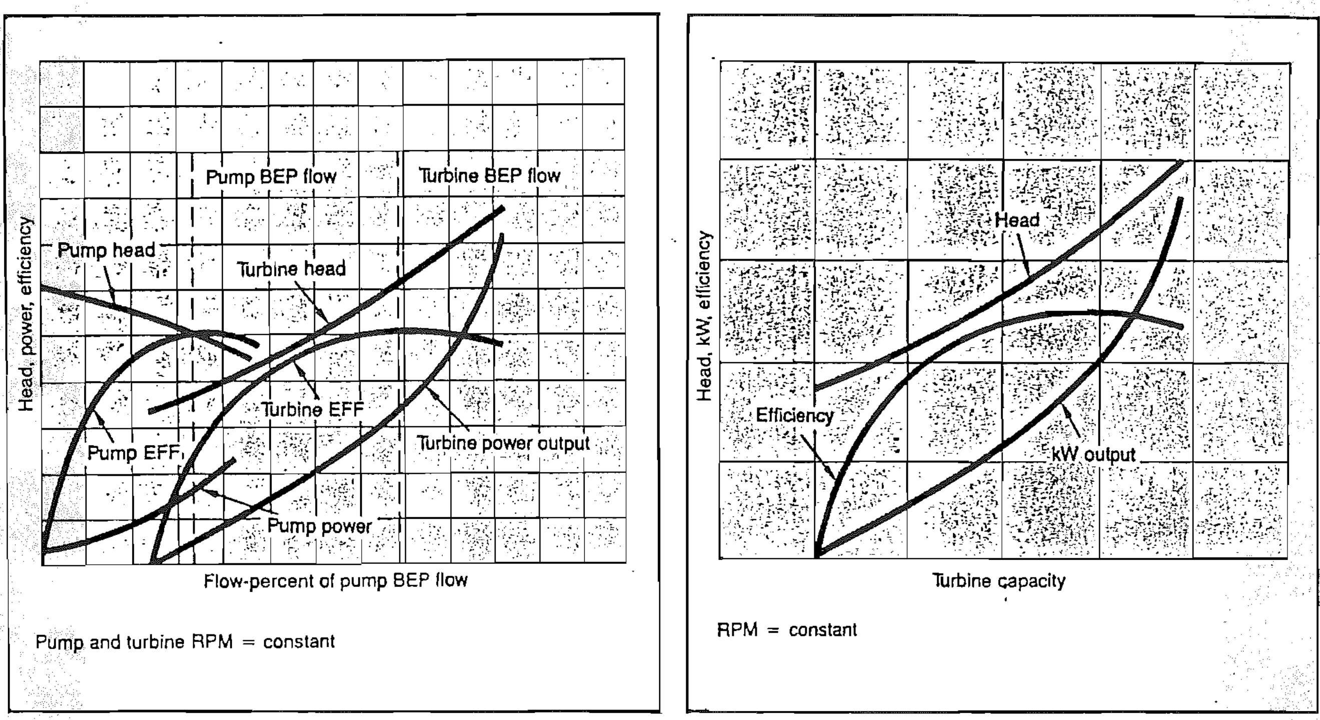


Fig 2.1 Normalized performance characteristics for a pump operating in the normal pump mode and in turbining mode

d) Turbine performance prediction

The biggest setback in choosing machines to operate as turbines from the vast range of available pumps is the lack of comprehensive turbine operating characteristics for the pumps over the range of specific speeds. Selection of a pump for use as a turbine, on the basis of pump data only, is difficult. However, there are various methods available for turbine performance prediction. Wong [58] has developed the following simple formulae relating pump operating characteristics to turbine operating characteristics.

$$H_p = H_t \eta_p \qquad \text{(E2.1)}$$

$$Q_p = Q_t \sqrt{\eta_p} \qquad \text{(E2.2)}$$

$$N_{sp} = N_{st} \sqrt{\eta_p} \qquad \text{(E2.3)}$$

On the other hand, Palgrave [53] uses velocity diagrams to determine from first principles simple correlations of pump and turbine performance. The method used requires comprehensive analysis and the exact geometry of the pump. There are other methods developed which look at dimensionless characteristic curves of pumps and various other pump design parameters to determine the equivalent turbining performance [16,19,27,37,39].

Despite the many methods available for turbine performance prediction, a turbine supplier is normally required to supply turbine mode, full-test data. However, acceptance and verification tests of pump as a turbine are expensive as compared with normal pump tests.

The turbining performance prediction method developed by Wong as described above represents a simple initial selection for full turbine testing or for initial pricing of plants.

2.4 GENERATORS

As outlined earlier in this chapter, the most important factor in the development of a microhydro system is the initial capital cost. The use of the electronic load governor significantly reduces the cost by eliminating the use of complex mechanical governors. It is envisaged by many researchers, that further cost reductions can be achieved by using induction motors in reverse mode as stand alone induction generators. There are some potential advantages in using an induction motor as a generator. The induction motor is cheap and rugged, and as an induction generator it is well suited to starting motors, but it has the problem of excitation and voltage control.

Synchronous alternating current machines on the other hand are more suited to stand-alone use, such as that encountered by isolated microhydro applications. A synchronous generator usually has a built-in excitation system and automatic voltage regulation. The suitability of synchronous generators for use with electronic load diverting governors is further analyzed in Chapter 6.

The economics of using induction generators for stand-alone application is not yet justified due to the high cost of producing the external excitation and voltage regulation systems.

2.5 ELECTRICAL SAFETY AND MALFUNCTION PROTECTION SYSTEM FOR MICROHYDRO PLANTS

Most microhydro applications exist in areas of high rainfall, high humidity mountainous terrain and lacking an electrical distribution system. In such cases, microhydro plants are vulnerable to possible malfunction, both electrically and mechanically.

A microhydro generating set principally comprises a water power system and an electrical system interconnected by a mechanical power transmission system. The electrical system itself comprises the generation, transmission and consumer reticulation. Control processes are most effectively handled using electrical components and in microhydro power the preference is as far as possible, to use electrical means to protect the whole scheme against malfunction of all kinds.

Bryce and Giddens [3] have developed a fail safe electrical malfunction protection system specifically for stand-alone microhydro plants. Their concept is analyzed further in Chapter 7.

The system's main purpose is to protect the consumer's appliances and the generating equipment from operating at an abnormal condition as a result of malfunctions in any part of the complete scheme or the inadvertent misuse of the installation by the consumer. The malfunction protection system simply applies the brake and shuts down the plant when voltage, current and frequency deviate from the set range.

The function of the load management system is to allow the consumer full use of the available power without the risk of causing damage to the plant from overloading, as in most cases the design output of the generator will normally be much less than the load which can be connected to it by the consumer. Similarly, the mechanical power to the generator may fall below the design value due to a reduction in water supply with the same consequent overloading effect. This situation does not call for immediate isolation, but for the consumer to make a load adjustment. An alarm warning of the need to shed load, is followed by isolation after a short time interval.

CHAPTER THREE

3 BACKGROUND HYDRAULIC THEORY OF MICROHYDRO SYSTEMS

3.1 PRINCIPLES OF FLUID MACHINERY

Fluid machines either add or subtract energy from the working fluid and in many cases the process is reversible using the same machines, i.e. machines that add energy to the fluid can be used to subtract energy from the fluid and vice versa.

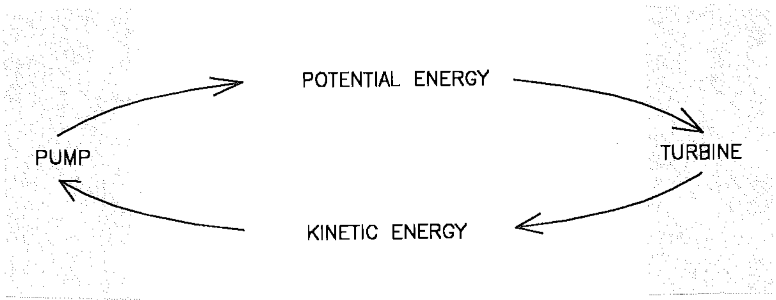


Fig 3.1 Energy flux of fluid machinery

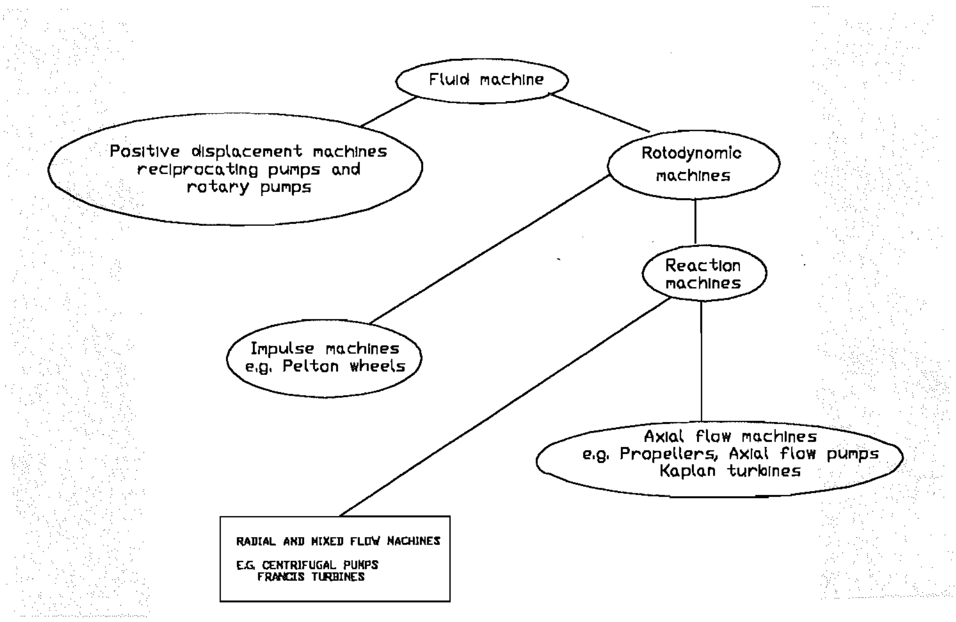


Fig 3.2 Classification of fluid machines

3.2 ROTODYNAMIC MACHINES

Rotodynamic machines are classified as either impulse or reaction types. The impulse machines operate at atmospheric pressure with jets flowing over moving vanes whereas the reaction machines run with rotating elements fully submerged and at pressures other than atmospheric. For any particular application, the reaction machine would run faster and be smaller in size than the equivalent impulse machine. This design is concerned with reaction machines.

If the machine is designed such that the energy transfer takes place while the fluid moves generally, radially through the impeller or runner, it is referred to as a radial flow reaction machine. On the other hand, if the energy transfer takes place while the fluid flows generally in the axial direction it is referred to as an axial flow reaction machine.

The centrifugal pump and Francis turbine are examples of radial flow reaction machines. Propellers, Kaplan turbines and axial flow pumps are axial flow reaction machines. Mixed flow machines combine features of both types and provide energy exchange while the fluid flows partially radially and partially axially. The result is the achievement of combined characteristics of radial flow (high head) and axial flow (low head) machines.

3.3 NET HEAD ACROSS A REACTION TURBINE

The effective head across any turbine is the difference between the head at the inlet to the machine and the head at the outlet from it.

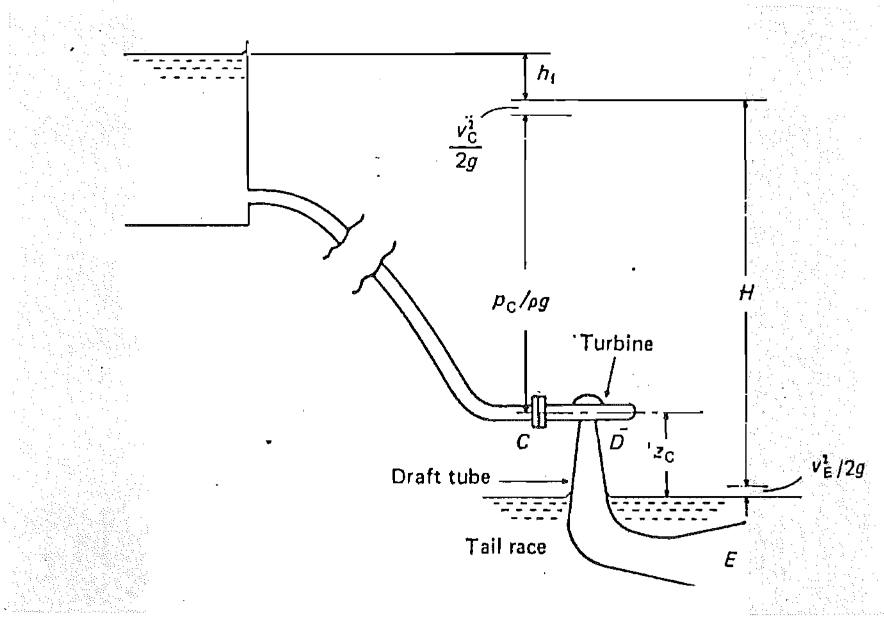


Fig 3.3 A typical hydro-turbine installation

The net head across the machine corresponds to the difference in the water level between the intake and the tail water, minus the losses external to the machine (that is the losses due to pipe friction and the head loss at outlet from the draft tube). Fig 3.3 indicates that the net head across the turbine is:-

H = Total head at inlet to the machines - Total head at discharge to tail water

$$H = \frac{P_c}{\rho g} + \frac{v_c^2}{2g} + Z_c - \frac{v_E^2}{2g} \qquad \text{(E3.1)}$$

a) Draft tube

As seen in Fig 3.3, a draft tube is fitted to the turbine. The function of the draft tube is to recuperate the kinetic energy remaining in the flow by reducing the velocity of the water leaving the turbine, that is to recuperate the $v^2/2g$ component, and so important is this function of the draft tube that it is usually considered as part of the turbine.

The pressure at outlet from the impeller (runner), i.e. inlet to the draft tube is usually less than atmospheric. The turbine should therefore not be set so high above the tail water (ie large suction head) that the pressure at the outlet from the impeller falls to such a low value that cavitation occurs (see Section 3.6).

3.4 BASIC EQUATION OF ROTODYNAMIC MACHINERY

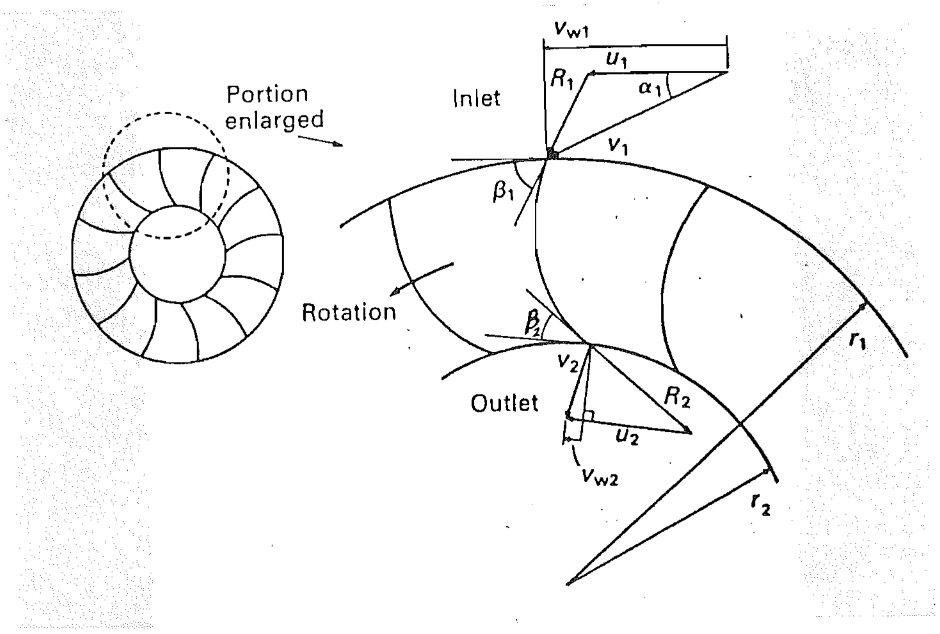


Fig 3.4 Velocity diagrams of rotodynamic pumps and turbines

The equation of continuity, the momentum equation and the general energy equation are used, but certain special forms of these relationships are developed.

v	Absolute velocity of fluid
u	Peripheral velocity of blade at point considered
R	Relative velocity between fluid and blade
v_w	Velocity of whirl ie. component of absolute velocity of fluid in direction tangential to runner circumference
r	Radius of runner
ω	Angular velocity of runner
Suffix *1*	Condition at inlet
Suffix 2	Condition at outlet

The only movement of the runner blades is in the circumferential direction and so only force components in this direction perform work.

Torque about a given fixed axis is equal to the rate of increase of angular momentum about that axis

The torque on the fluid must be equal to the angular momentum of the fluid leaving the rotor per unit time minus the angular momentum of the fluid entering the rotor per unit time.

At inlet, any small particle of fluid of mass δm has momentum $\delta m v_{w1}$ in the direction tangential to the rotor.

Angular momentum (moment of momentum) is therefore,

$$\delta m v_{w1} r_1 \tag{E3.2}$$

Suppose that of the total (constant) mass flow rate m, a part δm passes through the small element of inlet cross section in which values of v_{w1} and r_1 are uniform. Then the rate at which angular momentum passes through the small element of inlet cross section is,

$$\int v_{w1} r_1 d\dot{m} \tag{E3.3}$$

The integral is taken over the entire cross section.

Similarly, the total rate at which angular momentum leaves the rotor is,

$$\int v_{w2} r_2 d\dot{m} \tag{E3.4}$$

and again the integral is taken over the entire cross section.

The rate of increase of angular momentum of the fluid is therefore,

$$\int v_{w2} r_2 d\dot{m} - \int v_{w1} r_1 d\dot{m} \tag{E3.5}$$

and this equals the torque exerted on the fluid.

If there are no shear forces at either inlet or outlet cross section which have a momentum about the axis of the rotor, then this torque on the fluid must be exerted by the rotor. By Newton's Third law a change of sign gives the torque exerted on the rotor by the fluid.

$$T = \int v_{w1} r_1 d\dot{m} - \int v_{w2} r_2 d\dot{m} \tag{E3.6}$$

(E3.6) is known as Euler's Equation. It is a fundamental relationship for all forms of rotodynamic machinery - turbines, pumps, fans or compressors.

The torque available from the shaft of the turbine is somewhat less than that given by (E3.6) because of friction in bearings and friction between the runner and the fluid outside it.

For a rotor, the shaft work done in unit time interval is,

$$T\omega = \int v_{w1}\omega r_1 d\dot{m} - \int v_{w2}\omega r_2 d\dot{m} \tag{E3.7}$$

Since u=ωr

$$T\omega = \int u_1 v_{w1} d\dot{m} - \int u_2 v_{w2} d\dot{m} \tag{E3.8}$$

The shaft work done by the fluid per unit mass is obtained by dividing (E3.8) by the total mass flowrate $\dot{m}$, then:-

Work done per unit mass of fluid,

$$\frac{1}{\dot{m}}\left(\int u_1 v_{w1} d\dot{m} - \int u_2 v_{w2} d\dot{m}\right) \tag{E3.9}$$

The integral (E3.9) can in general be evaluated only if it is known in what way the velocity varies over the inlet and outlet cross sections of the rotor. However, a simple result is obtained if the product $v_w r$ is constant at each cross section concerned. This assumption would be realistic if also the number of vanes guiding the fluid onto the rotor and the number of blades in the rotor were large so that there would be no significant variation of either inlet or outlet values of v_w with angular position.

In such a case (E3.9) becomes,

Work done per unit mass,

$$\frac{T\omega}{m} = u_1 v_{w1} - u_2 v_{w2} \tag{E3.10}$$

To the turbine, the energy available per unit mass of fluid is gH where H = net head.

Therefore, the hydraulic efficiency of the turbine now is,

$$\frac{(u_1 v_{w1} - u_2 v_{w2})}{gH} \tag{E3.11}$$

This represents the efficiency with which energy is transferred from the fluid to the rotor. It should be distinguished from the overall efficiency of the machine because, owing to such losses as friction in bearings and elsewhere, not all the energy received by the runner is available at the output shaft.

Not all the energy of the fluid is used by the turbine runner. The unused remaining energy is principally in the form of kinetic energy, and so, for high efficiency, the kinetic energy of the fluid at outlet should be minimal.

An alternative form of (E3.10) assumes uniform velocities at both the inlet and outlet. Given this assumption and application to Fig 3.4 it can be deduced that:-

$$R^2=u_1^2+v_1^2-2u_1v_1\cos\alpha_1=u_1^2+v_1^2-2u_1v_{w1} \qquad \text{(E3.12)}$$

$$\therefore u_1v_{w1}=\frac{1}{2}(u_1^2+v_1^2-R_1^2) \qquad \text{(E3.13)}$$

$$u_2v_{w2}=\frac{1}{2}(u_2^2+v_2^2-R_2^2) \qquad \text{(E3.14)}$$

Work done per unit mass,

$$\frac{1}{2}[(v_1^2-v_2^2)+(u_1^2-u_2^2)-(R_1^2-R_2^2)] \qquad \text{(E3.15)}$$

For turbines, work is done by the fluid, and since the flow passage decreases rather than increases in cross sectional area, (E3.15) must be positive.

i.e. $v_1 > v_2$
$u_1 > u_2$
$R_2 > R_1$

Similar relationships apply to pumps. There, the transfer of energy is from the rotor to the fluid instead of from the fluid to the rotor. So the (E3.15) is negative. For centrifugal pumps (E3.9) now becomes:-

Work done on fluid per unit mass,

$$\frac{T\omega}{m} = u_2v_{w2}-u_1v_{w1} \qquad \text{(E3.16)}$$

and (E3.15) becomes:-
Work done per unit mass,

$$\frac{1}{2}[(v_2^2-v_1^2)+(u_2^2-u_1^2)-(R_2^2-R_1^2)] \qquad \text{(E3.17)}$$

3.5 SIMILARITY LAW AND SPECIFIC SPEEDS

a) Specific speed

Specific speed is a very useful parameter for engineers involved in designing rotodynamic machines. In practical terms, specific speed is a tool for use in comparing various rotodynamic machines and then selecting the most efficient and economic equipment for the application.

Specific speed, sometimes referred to simply as a type number, is defined as the reference number that describes the hydraulic features of the machine. Specific speed is always calculated at the best efficiency point (BEP). For geometrically similar machines ie. completely scale-up or down, and with complete similarity of flow (i.e. same density of liquid) specific speed remains unchanged.

The common expressions for specific speeds are as follows. They are of dimensionless form.

For pumps,

$$N_{sp} = N\frac{\sqrt{Q}}{(gH)^{3/4}} \qquad \text{(E3.18)}$$

Where Q is in m^3/s.

For turbines,

$$N_{st} = N\frac{\sqrt{P/\rho}}{(gH)^{5/4}} \qquad \text{(E3.19)}$$

When the site of the installation and the output required are known, the specific speed may be calculated and the type of machines best suited to these conditions selected. For the principle types of turbine, the range of specific speeds is shown in Fig 3.5.

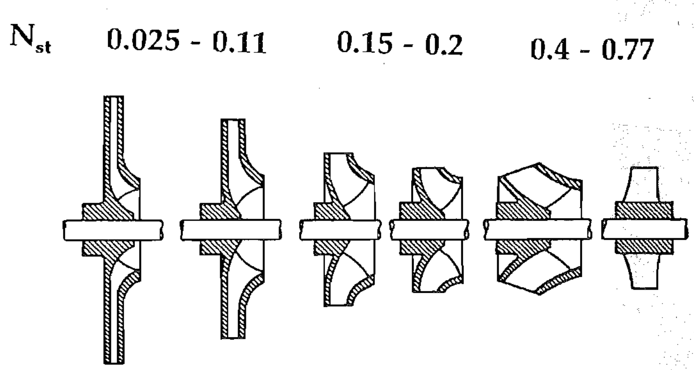


Fig 3.5 Dimensionless specific speeds with different shape impellers

Fig 3.5 indicates the variation of specific speed with the shape or the turbine runner. The value quoted should be regarded as approximate because of the shapes of the other parts of the turbines. For example, the volute, the guide passage and the draft tube affect the specific speed to some extent.

b) Affinity law (similarity law)

The well known affinity laws establish relationships between the performance parameters of geometrically similar machines operating under conditions of kinematic similarity. Dynamic similarity would require similarity of all forces but Reynolds number similarity is impractical and the commonly used similarity laws ignores changes in Reynolds number. The consequence of this is that a scale effect is introduced and efficiency remains unchanged between, different but, otherwise similar, cases.

The basic rules are:-

$$v_r = N_r D_r$$

$$Q_r = v_r D_r^2 = N_r D_r^3$$

$$H_r = v_r^2 = N_r^2 D_r^2$$

$$P_r = Q_r H_r \rho = N_r^3 D_r^5$$

$$\eta_r = 1$$

Where, subscript r denotes ratios of the quantities.

A separate but related consideration is the trimming of the machine impeller to reduce performance. This is a common practice in pumping applications but experience is necessary for relevance to pumps in the turbining mode.

The trimming of the impeller does change the geometry of the pump but it has been found that for small changes in impeller diameter, another set of rules can be applied for pumping applications. These are, that for the same speed of operation,

$$Q_r = D_r$$
$$H_r = D_r^2$$

It is also not unusual for a small change in machine speed to be considered at the same time as a change in impeller diameter. The effect of speed change, by itself, depends on the orthodox similarity rules. Thus, for this particular procedure - making small adjustments to impeller diameter and machine speed to fulfil a particular required duty - the two functions can be considered together and are set out in Table 3.1. [42,57]

Diameter change only, constant speed	Speed change only, constant diameter	Diameter and speed change
$Q_r \quad = D_r$	$Q_r \quad = N_r$	$Q_r \quad = D_r N_r$
$H_r \quad = D_r^2$	$H_r \quad = N_r^2$	$H_r \quad = D_r^2 N_r^2$
$P_r \quad = D_r^3$	$P_r \quad = N_r^3$	$P_r \quad = D_r^3 N_r^3$

Table 3.1 Formulae for refiguring reaction machine performance with impeller diameter or speed change [42].

3.6 CAVITATION

In a liquid, if the pressure at any point falls to the vapour pressure (at the temperature concerned), the liquid boils and small bubbles of vapour form in large numbers. These bubbles are carried along by the flow, and on reaching a point where the pressure is higher they suddenly collapse as the vapour condenses to liquid again. The surrounding liquid rushes in to fill the bubble cavities. The liquid moving from all directions collides at the centre of cavity, thus giving rise to a very high, local pressure (up to 1 GPa). Any solid surface in the vicinity is subjected to these intense pressures, because even if the cavities are not actually at the solid surface, the pressures are transmitted from the cavities by pressure waves similar to those encountered in water hammer (see Section 8.4.c). This alternate formation and collapse of vapour bubbles may be repeated with a frequency of many thousand times per second. The intense pressure, even though acting for a very short interval over a tiny area can, with repetitions, cause severe damage to the surface. This phenomenon is called CAVITATION.

Associated with cavitation may be considerable vibration and noise and when cavitation occurs in a turbine or pump, it may sound as though gravel were passing through the machine.

In reaction turbines (such as centrifugal pumps in reverse) the point of minimum pressure is usually at the outlet end of the impeller, on the leading side.

For the flow between such a point and the final discharge into the tail race (when the total head is atmospheric), the energy equation may be written as:-

$$\frac{P_{\min}}{\rho g} + \frac{v^2}{2g} + z - H_f = \frac{P_{atm}}{\rho g} \qquad \text{(E3.20)}$$

Where H_f represents head loss due to friction in the draft tube, and P_{min} is the minimum absolute total pressure required at the turbine suction nozzle for cavitation not to occur.

3.7 PREVENTING CAVITATION IN PUMPING

a) Definition of net positive suction head (NPSH)

The net positive suction head is the amount by which the absolute total head at the pump suction nozzle, exceeds the absolute vapour pressure head of the flow. The need for such an excess is due to local reductions in pressure in the impeller inlet, resulting from increases in local velocities, compared with pressure at the pump suction nozzle where measurements can normally be taken.

Pumps tests show that a minimum value of the NPSH is required to avoid cavitation. The NPSH characteristic of a pump shows values of the required NPSH (i.e. the NPSHR), necessary to avoid cavitation, for a range of operating conditions.

To consider the appropriateness of the site for a pump, the NPSH that is available at the site (i.e. the NPSHA) can be determined independently of the NPSHR. It is then necessary that the NPSHA is not less than the NPSHR if cavitation is to be avoided.

By definition, NPSHA is,

NPSHA = ***Absolute total head***
- Absolute vapour pressure head
- Line loss
$+ \Delta z$

It should be noted that all pressure heads are referenced to the pump suction nozzle where measurements can be taken.

3.8 PREVENTING CAVITATION IN TURBINING

Just as pumping requires a minimum NPSH, turbine duties require net positive exhaust heads known as Total Required Exhaust Head (TREH). The total required exhaust head is the amount by which the absolute total head at the turbine exit or suction nozzle (i.e. analogous of pump suction nozzle) exceeds the absolute vapour pressure head of the fluid [4]. Rearrange (E3.20) to give,

$$\frac{v^2}{2g} - h_f = \frac{P_{atm}}{\rho g} - \frac{P_{\min}}{\rho g} - z \qquad \text{(E3.21)}$$

For a particular design of machine operated under its design conditions, the left hand side of this relationship may be regarded as a particular proportion, say σ_c, of the net head H across the machine. Then, critical cavitation constant is defined as,

$$\sigma_c = \frac{(P_{atm}/\rho g) - (P_{\min}/\rho g) - z}{H} \qquad \text{(E3.22)}$$

The top portion of E3.22 is the TREH. It then follows that,

$$\sigma_c = \frac{TREH}{H} \qquad \text{(E3.23)}$$

For cavitation not to occur, p_{min} must be greater than absolute vapour pressure of the liquid p_v.

In practice the expression is used to determine the maximum elevation z of the turbine above the tail water surface for cavitation to be avoided.

Rearrange (E3.22) to give:-

$$z_{max} = \frac{P_{atm}}{\rho g} - \frac{P_v}{\rho g} - \sigma_c H \qquad \text{(E3.24)}$$

E3.24 indicates that the greater the net head H on which a given turbine operates, the lower it must be placed relative to the tail water level. i.e. small Δz.

3.9 DETERMINING THE CRITICAL CAVITATION CONSTANT

Cavitation parameters can be derived from Fig 3.6 [45]. The figure indicates that a turbine of high specific speed has a higher value of σ_c (i.e. approaching axial flow for reaction machines) and so they must be set much lower than those of smaller specific speed.

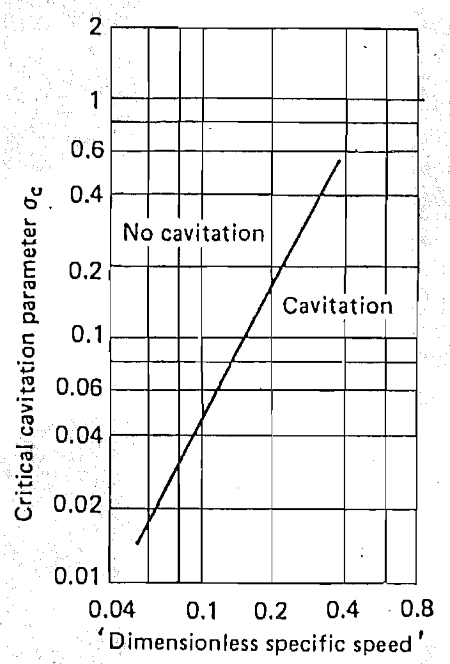


Fig 3.6 Cavitation limits for Francis Turbines

CHAPTER FOUR

4 TURBINE SELECTION AND TESTING

4.1 DEFINITION OF RANGE OF TURBINES

There are many advantages of using centrifugal pumps as turbines for microhydro systems as outlined in Chapter 2. Stepanoff, Buse and various others [57,4], have proven that centrifugal pumps can operate very effectively as turbines with good efficiency characteristics. There are various methods of estimating turbine performance from pump performance data, although, these methods do not give an accurate assessment of the turbine performance characteristics, and vary from pump to pump depending on the design. Actual testing is required on each machine to determine its turbining performance characteristics.

This chapter outlines the results from turbine tests carried out at the Fluid Mechanics Laboratory of the Civil Engineering Department, University of Canterbury. The turbine testing results to be presented are analysed leading to the formulation of a simple design procedure for specifying performance of a range of ISO standard centrifugal pumps as turbines. The final step is the construction of a selection chart which would enable the microhydro plant developer to select an appropriate pump that matches the hydro site conditions with minimal effort.

4.2 INITIAL TURBINE SELECTION

Centrifugal pumps manufactured in Australia by Thompsons, Kelly and Lewis Ltd, were selected for the purpose of this exercise based on their ruggedness and their design to ISO standard specifications. The five pumps selected are shown in Fig 4.1. Full detail specification of these pumps is outlined in Appendix 1 and Appendix 5.

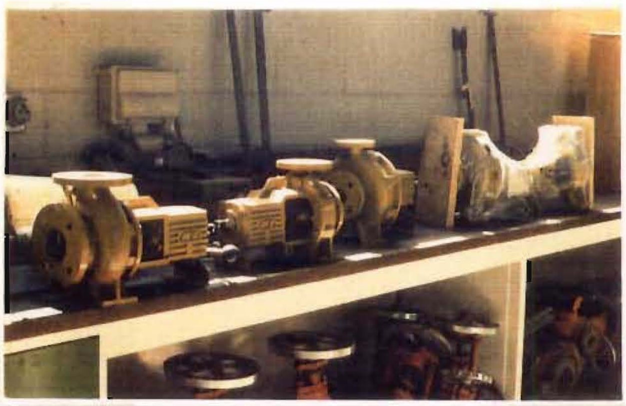


Fig 4.1 The KL-ISO Centrifugal pumps

The pumps were selected initially based on the following turbining guidelines :-

1 6 kW shaft power output.
2 Head and discharge ratio of turbine to pump of 1.5 to 2.
3 Turbine efficiency equals 98% of pump efficiency.
4 Similarity rules without scale effect.

From these assumptions the 5 pumps of the KL-ISO range were selected:-

1 KL-ISO 50 x 32 - 200
2 KL-ISO 65 x 50 - 200
3 KL-ISO 65 x 50 - 160
4 KL-ISO 80 x 65 - 160
5 KL-ISO 80 x 65 - 125

KL-ISO 50 x 32 - 200 for example:

50 pump inlet diameter or turbine discharge diameter (mm)
32 pump discharge diameter or turbine inlet diameter (mm)
200 nominal impeller diameter (mm)

A 6 kW shaft power output would give 4.8 kW of electrical output at 0.8 power factor. (See section 5.1). The pumps were tested both in their conventional pumping mode and in the reversed turbining mode.

4.3 TEST METHOD AND PROCEDURE

The tests were conducted with normal standard arrangements to furnish and measure the water supply and a variable speed electric dynamometer was used to measure the shaft power.

The following parameters were to be determined:-

a) Turbine test

1	Head (m)	Differential total head
2	Discharge (l/s)	Determined from magnetic flow meter
3	Power (kW)	Output shaft power determined from the dynamometer
4	Efficiency (%)	Turbine efficiency is the ratio of output shaft power to the available water power ($\eta = P/\rho gQH$)

b) **Pump test**

1	Head (m)	Differential total head
2	Discharge (l/s)	Determined from magnetic flowmeter
3	Power (kW)	Input shaft power determine from the dynamometer
4	Efficiency (%)	Pump efficiency is the ratio of the delivered water power to the input shaft power ($\eta = \rho gQH/P$)

4.4 THE PUMP/TURBINE TEST RIG

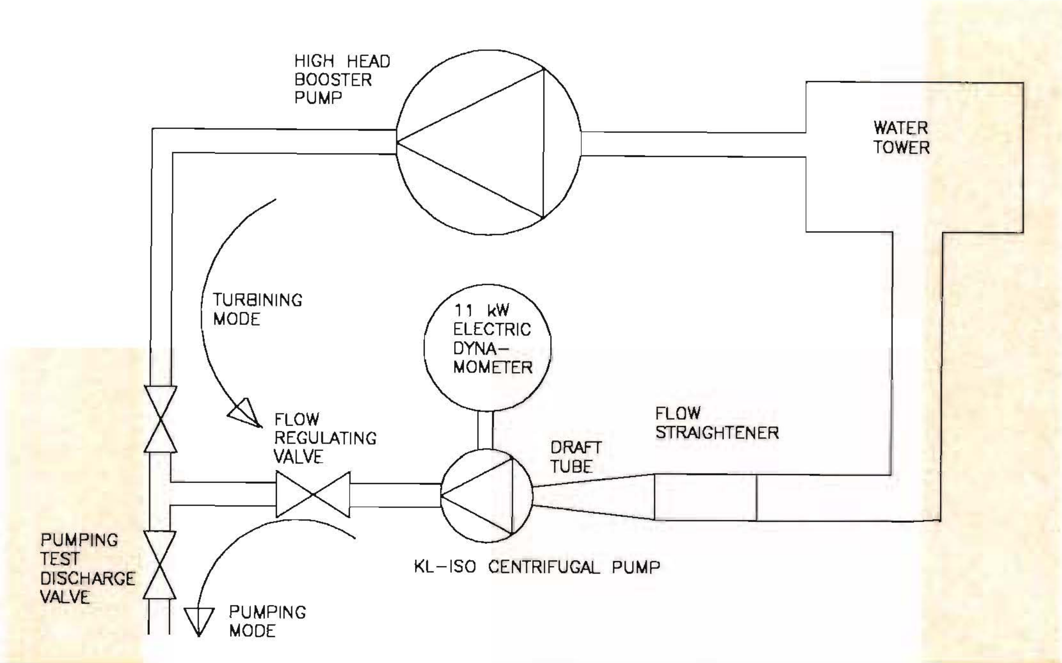


Fig 4.2 Schematic layout of the pump and turbine test rig

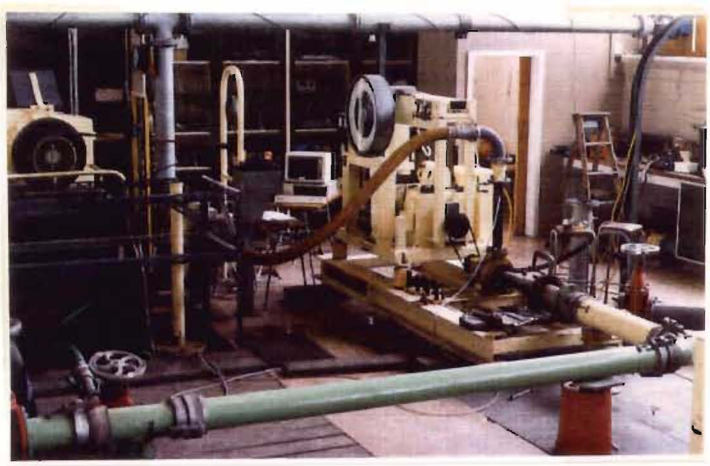


Fig 4.3 The pump/turbine test rig

Simple guidelines, based on empirical evidence with earlier installations of microhydro, were used to assist the experimental set-up as shown in Fig 4.2 and 4.3. When centrifugal pumps operate in the turbining mode, it is expected that at the same speed of rotation the turbining case will require a greater head and flow rate at the point of maximum efficiency than for the pumping case as outlined in Chapter 2. This supports the use of large diameter pipework with the appropriate contracting and enlarging fittings provided at the inlet and outlet.

In the turbining case, the utilisation of the energy of the site can be maximised by recuperating the kinetic energy of the water leaving the turbine with a draft tube as explained in chapter 2. The water leaving the machine is likely to have a swirl component which hinders the identification of the integrated total energy of the flow leaving the machine. For these reasons, the machine was tested with a draft tube comprising a conical diffuser and a flow straightener.

The measurement of the total energy of the outlet flow, when turbining, was made after the flow straightener. The draft tube was considered to be part of the turbine. In the pumping case, the water total energy was measured at the same position when the draft tube was operating in reversed flow direction. The draft tube provided an excellent flow entry condition for the pumping operation.

The head was measured using Budenberg Standard Test pressure gauges with 150 mm diameter face and with span of 0 to 200 psi (0 to 1.4 MPa). The signal lines were flushed and vented with allowances being made for the gauge height and velocity heads. The pressure tapping was properly formed in accordance BS 5316 Part 2 1977. Adequate lengths of straight pipe were provided for the flow approaching the pressure tappings. An allowance was made also for the small friction loss in the pump delivery pipe (turbine inlet pipe) and these results were corrected to the flange.

The discharge was measured with an 80 mm Fisher and Porter magnetic flow-meter, which had been checked and calibrated, and used in accordance with the manufacturer's instruction.

The shaft power was measured with an 11 kW, AC, electric dynamometer that could be used in the driving and absorbing mode. Before use, it was calibrated for torque measured by dead weight loading at the test speed of rotation. Because the maximum speed of the dynamometer was limited to 2500 RPM a timing belt drive with a speed ratio of 3:4 was used to test the pumps at the required speed of 3000 RPM. An allowance of 190 W was made, based on the advice obtained from the belt manufacturer for losses in the timing belt.

The speed was measured with an optical electronic hand-held tachometer.

To eliminate any problems of air penetration into the water system or from cavitation, all tests were conducted with a drowned suction having a positive gauge pressure of approximately 120 kPA.

4.5 ACCURACY OF THE PERFORMANCE DATA

The error in discharge is considered to be no more than +/- 1.0% of each individual reading based on evidence stated in the manufacturer's catalogue.

The error in the total head is considered to be no more than +/- 0.5 m for any reading based on Budenberg standard test gauges.

The error in shaft power is considered no more than +/- 100 W based on the calibration conducted on the dynamometer.

Overall, the efficiency deduced from the above data is considered to be in error by no more than +/- 1% for any individual result.

4.6 PUMP AND TURBINE PERFORMANCE CURVES

The pump performance curves for the five machines selected are shown in Figs 4.4 to 4.8 inclusive. The curves were compatible with the manufacturer's catalogue. Figs 4.9 to 4.15 inclusive show the corresponding turbine performance.

Comparison between pump performance curves and turbine performance curves shows the similarity between maximum efficiencies for both modes of operation. However, they occur at different discharges. The performance ratios at the best efficiency point for all the machines tested are outlined in Section 4.6.b.

a) Optimum operating condition

It was required that the turbine delivers 6 kW shaft power at the point of best efficiency. This was accomplished with three of the five pumps in the standard configuration. The other two pumps developed excessive power at the BEP and it was necessary to trim the impeller diameters to meet the specification. This was done in steps and test data obtained for the process of trimming the impeller diameter. The optimum operating points for the five turbines at 6 kW shaft power and 3000 rpm are shown in Fig 4.16. The effect of impeller trimming of the 80x65-160 machine is shown in Fig 4.17. Figs 4.17A and 4.17B show the impeller of the machine 80x65-160 with a new diameter of 142 mm.

Figs 4.17A, 4.17B A typical trimmed impeller (for machine 80x65-160[142])

For the above pumps, the testing programme also provided information on the turbining performance of each pump when operated at the same constant speed but under a reducing head. With decreasing head, the discharge reduced and the power output decreased. The efficiencies also reduced but not prohibitively so. The optimum operating conditions at 6 kW shaft power are shown in Table 4.1.

Machines	Head (m)	Discharge (l/s)	efficiency (%)
50x32-200	155	9	44
65x40-200[185]	85	13	56
65x50-160	64	14	70
80x65-160[142]	43	21	72
80x65-125	28	28	78

Table 4.1 Optimum operating conditions at 6 kW shaft power in turbining mode

The values in the square brackets represent actual diameter of the trimmed impeller.

b) Performance ratio

A useful outcome of the test data is information on how the ratios of head, discharge, power and efficiency between the turbining and pumping performance (for operating at the same speed at the BEP) vary with specific speed. The pumps shapes range from radial to mix flow. The trends in these ratios are shown in Table 4.2 and Figs 4.18 and 4.19. Note that the lines joining the points on the graphs can only indicate a trend. i.e. they have no physical meaning between the points because of the limited number of pumps available from the KL-ISO range.

Machines	N_{st}	N_{sp}	Head ratio	Disch. ratio	Effic. ratio	Power ratio
50x32-200	0.025	1.7	2.20	1.66	0.91	0.76
65x40-200[185]	0.050	2.5	1.76	1.28	0.98	0.88
65x50-160	0.075	3.7	1.57	1.23	1.03	0.96
80x65-160[142]	0.125	5.2	1.44	1.23	1.03	1.04
80x65-125	0.200	7.9	1.35	1.38	1.03	1.06

Table 4.2 Turbine to pump performance ratios at BEP

It should be noted that these results can only be applied with confidence to the pumps tested. The performance of small commercial pumps is likely to depend on manufacturing methods as well as fluid flows considerations. The results shown should be used with caution in considering pumps of the same make but different size and, particularly, when considering other makes of pumps.

4.7 TURBINE SELECTION CHART

The results for the performance of the five turbines are shown in Fig 4.20. Fig 4.21 also shows performance result but with the high head 50x32-200 turbine excluded to highlight the remaining four turbines. The two figures serve as selection charts. The lines joining the points on the graphs can only indicate a trend.

4.8 CONCLUSION

1 As a result of investigating the performance of KL-ISO pumps, a set of pumps has been identified that will, when operating as turbines, develop 6 kW for each of five different heads from 155 m to 28 m when operating at the best efficiency point at a speed of 3000 rpm. This fulfils the requirement of a range of pumps suitable for a standardised microhydro package to generate 5 kW at 230 v, 50 Hz, for a range of heads.

2 Information has been obtained on the performance of each turbine at 3000 rpm but under conditions of reduced head when the output reduces.

3 The effect of variation in specific speed on the ratios of the performance parameters between the pumping and turbining mode has been identified for this set of pumps.

Fig 4.4

KL-ISO CENTRIFUGAL PUMP
PUMP PERFORMANCE CURVE

KL-ISO 50x32-200
3000 RPM

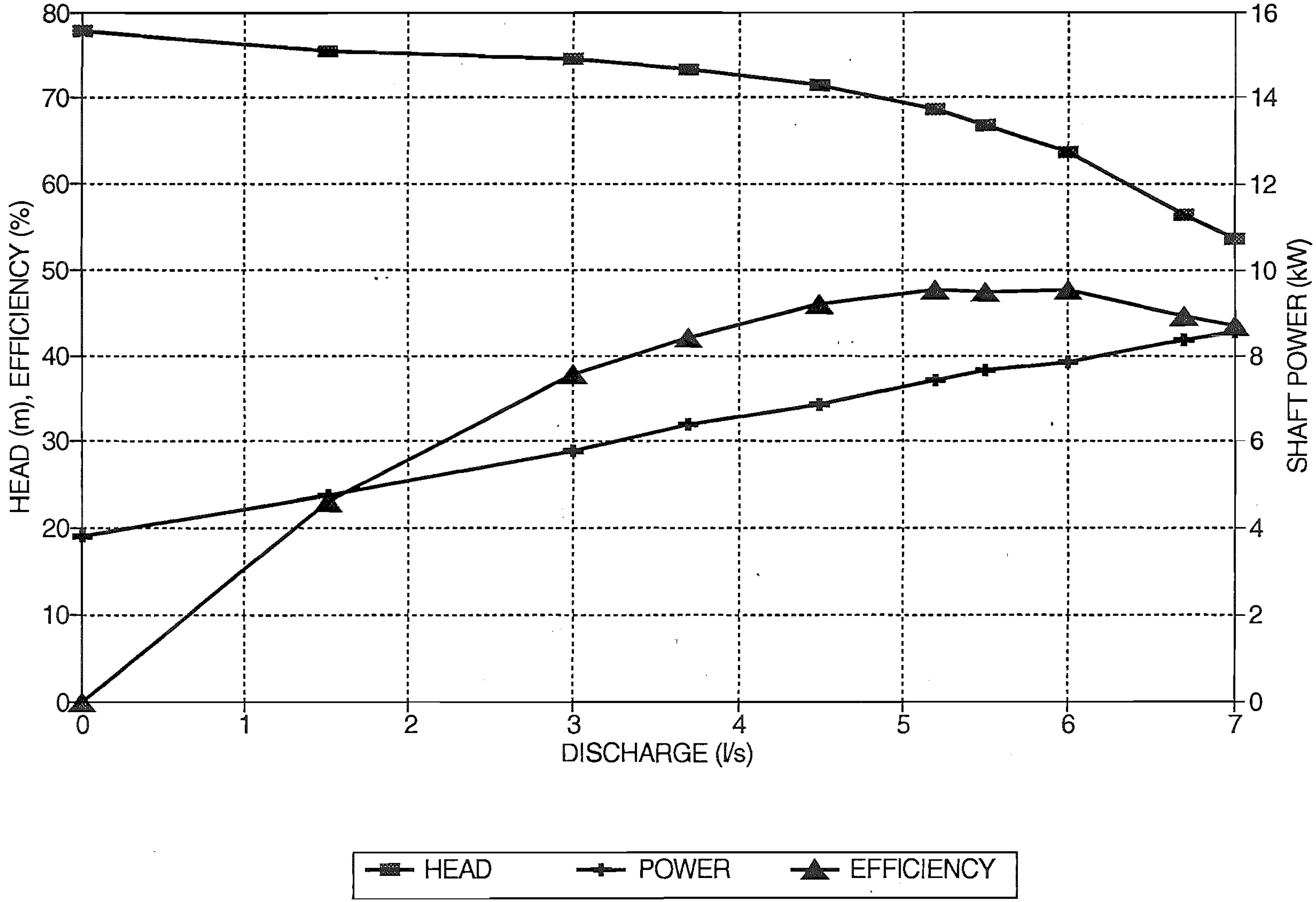
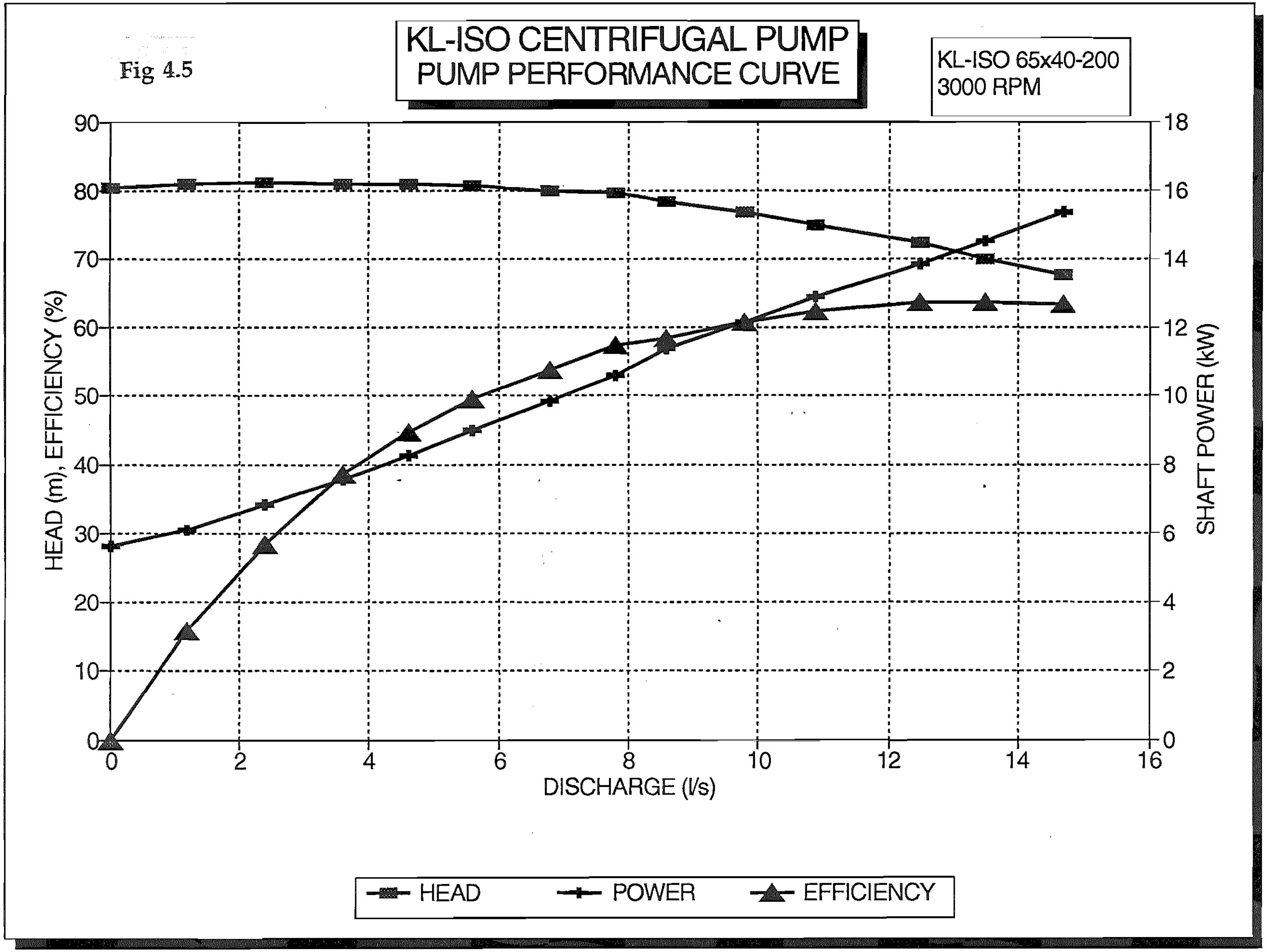


Fig 4.5



KL-ISO CENTRIFUGAL PUMP
PUMP PERFORMANCE CURVE
KL-ISO 65x40-200
3000 RPM
HEAD (m), EFFICIENCY (%)
SHAFT POWER (kW)
DISCHARGE (l/s)
HEAD
POWER
EFFICIENCY

Fig 4.6

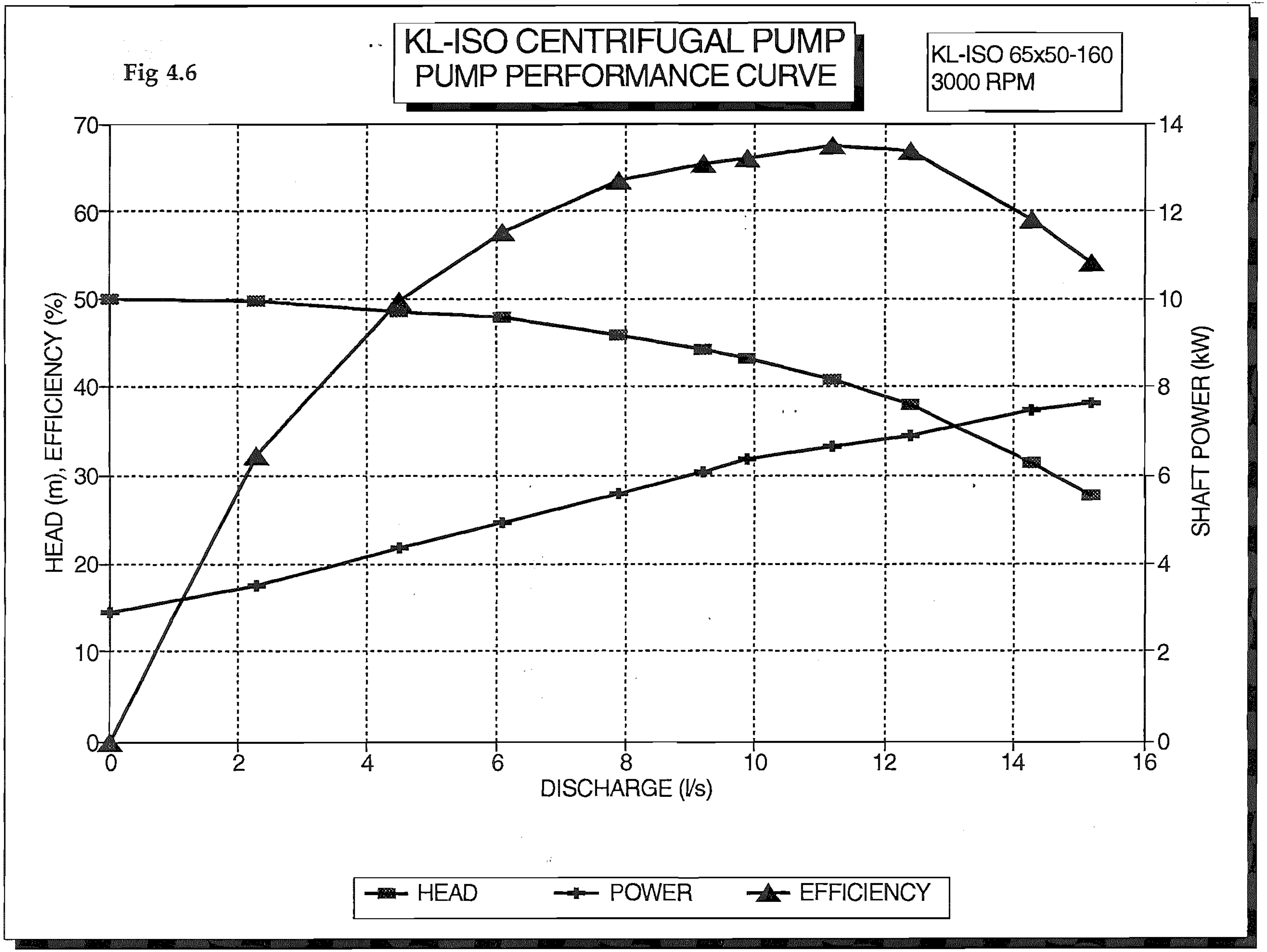

KL-ISO CENTRIFUGAL PUMP
PUMP PERFORMANCE CURVE
KL-ISO 65x50-160
3000 RPM
HEAD (m), EFFICIENCY (%)
SHAFT POWER (kW)
DISCHARGE (l/s)
0
10
20
30
40
50
60
70
0
2
4
6
8
10
12
14
16
HEAD
POWER
EFFICIENCY

Fig 4.7

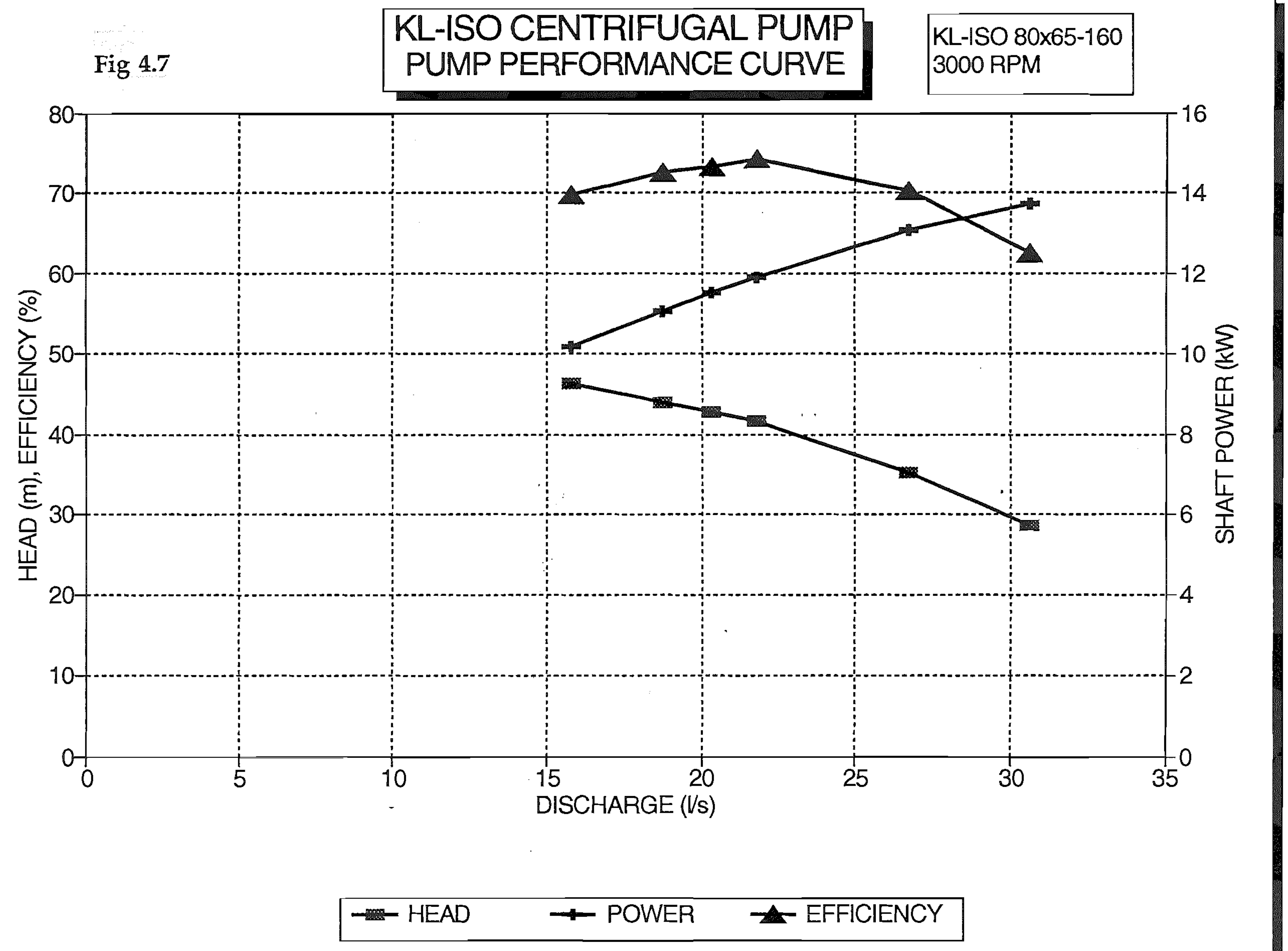

KL-ISO CENTRIFUGAL PUMP
PUMP PERFORMANCE CURVE
KL-ISO 80x65-160
3000 RPM
HEAD (m), EFFICIENCY (%)
SHAFT POWER (kW)
DISCHARGE (l/s)
0
10
20
30
40
50
60
70
80
0
2
4
6
8
10
12
14
16
0
5
10
15
20
25
30
35
HEAD
POWER
EFFICIENCY

Fig 4.8

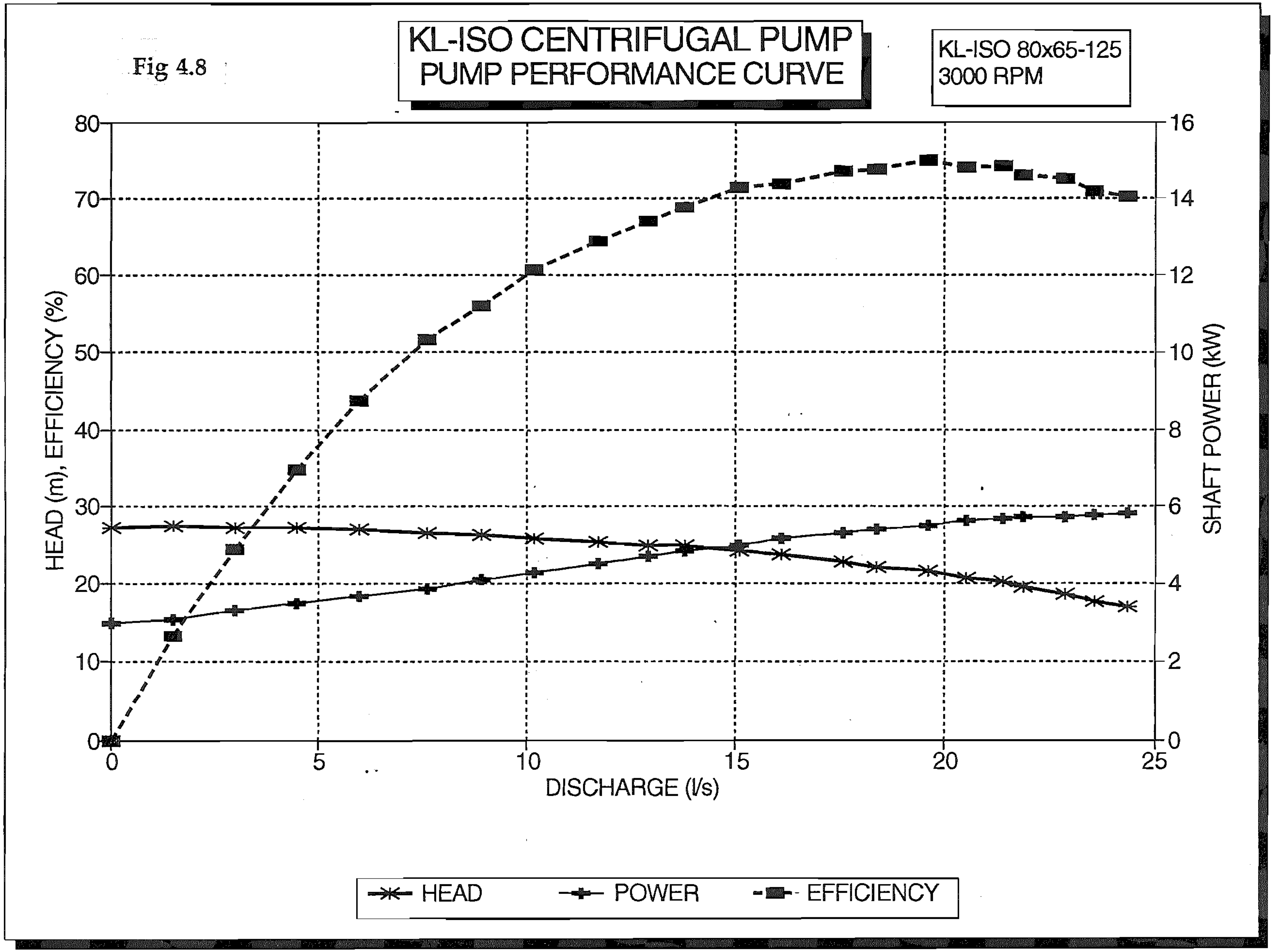
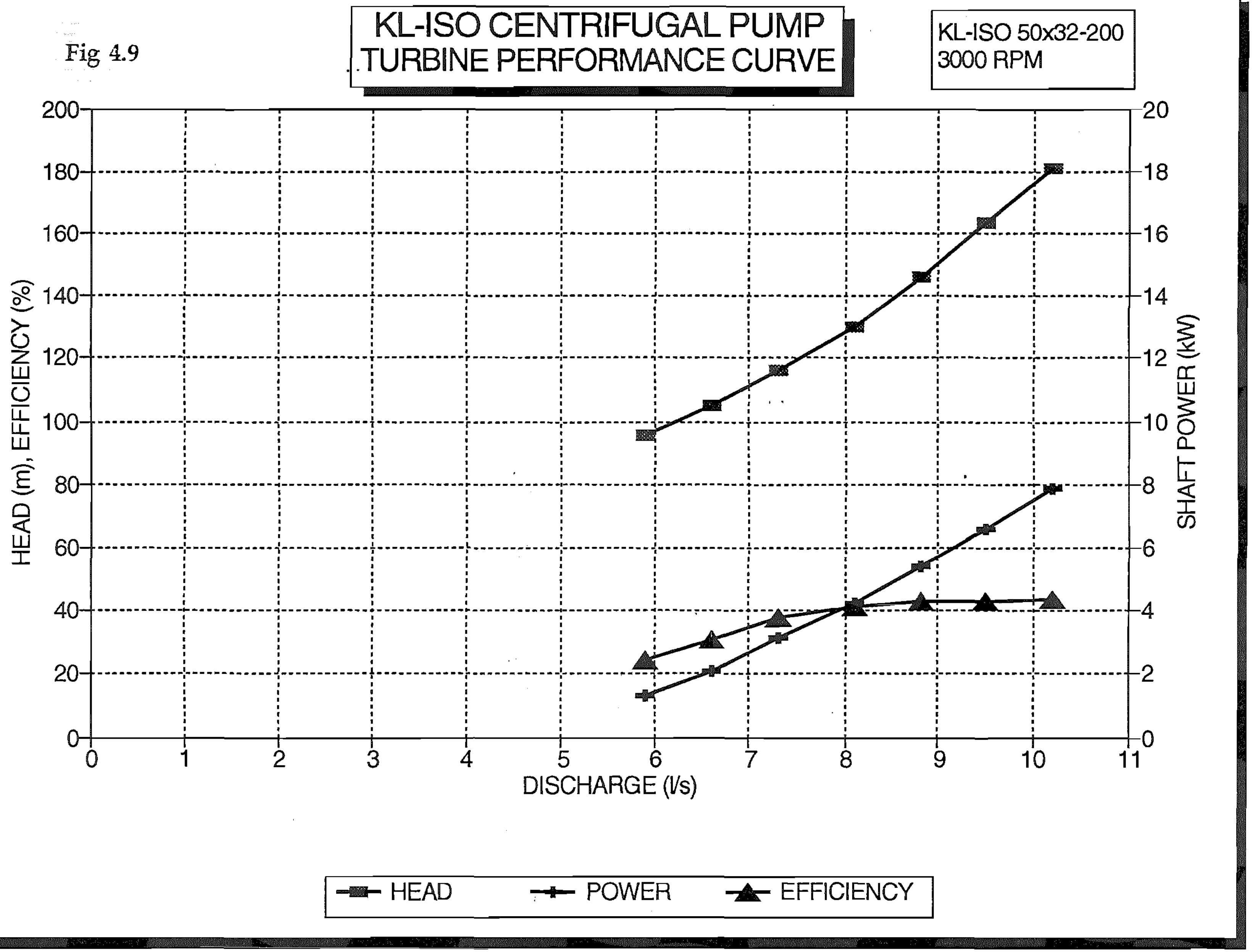
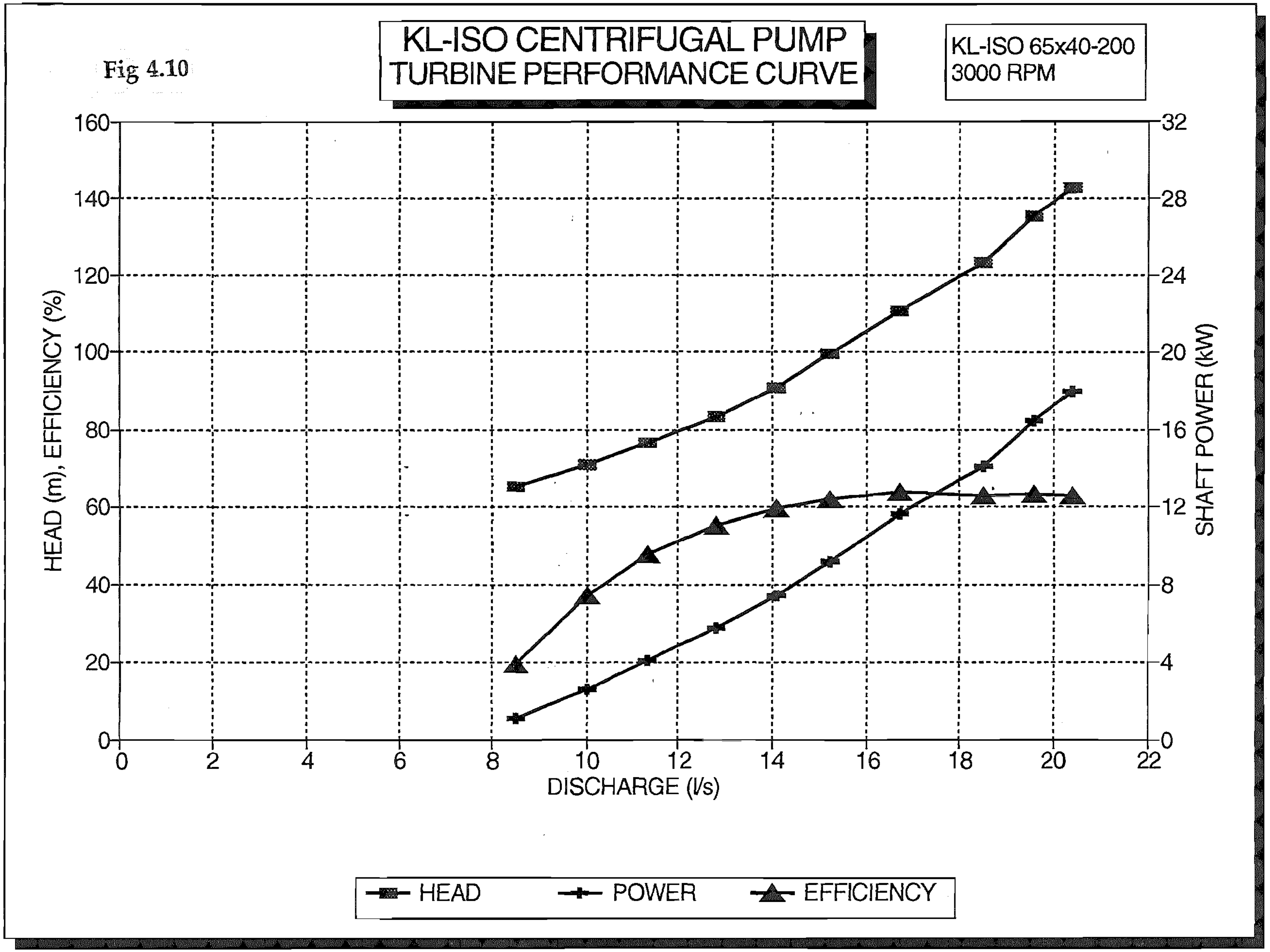

KL-ISO CENTRIFUGAL PUMP
PUMP PERFORMANCE CURVE
KL-ISO 80x65-125
3000 RPM
HEAD (m), EFFICIENCY (%)
SHAFT POWER (kW)
DISCHARGE (l/s)
0
10
20
30
40
50
60
70
80
0
2
4
6
8
10
12
14
16
0
5
10
15
20
25
HEAD
POWER
EFFICIENCY

Fig 4.9



KL-ISO CENTRIFUGAL PUMP
TURBINE PERFORMANCE CURVE
KL-ISO 50x32-200
3000 RPM
HEAD (m), EFFICIENCY (%)
SHAFT POWER (kW)
DISCHARGE (l/s)
0
1
2
3
4
5
6
7
8
9
10
11
0
20
40
60
80
100
120
140
160
180
200
0
2
4
6
8
10
12
14
16
18
20
HEAD
POWER
EFFICIENCY

Fig 4.10



KL-ISO CENTRIFUGAL PUMP
TURBINE PERFORMANCE CURVE
KL-ISO 65x40-200
3000 RPM
HEAD (m), EFFICIENCY (%)
SHAFT POWER (kW)
DISCHARGE (l/s)
0
2
4
6
8
10
12
14
16
18
20
22
20
40
60
80
100
120
140
160
28
24
16
12
32
HEAD
POWER
EFFICIENCY

Fig 4.11

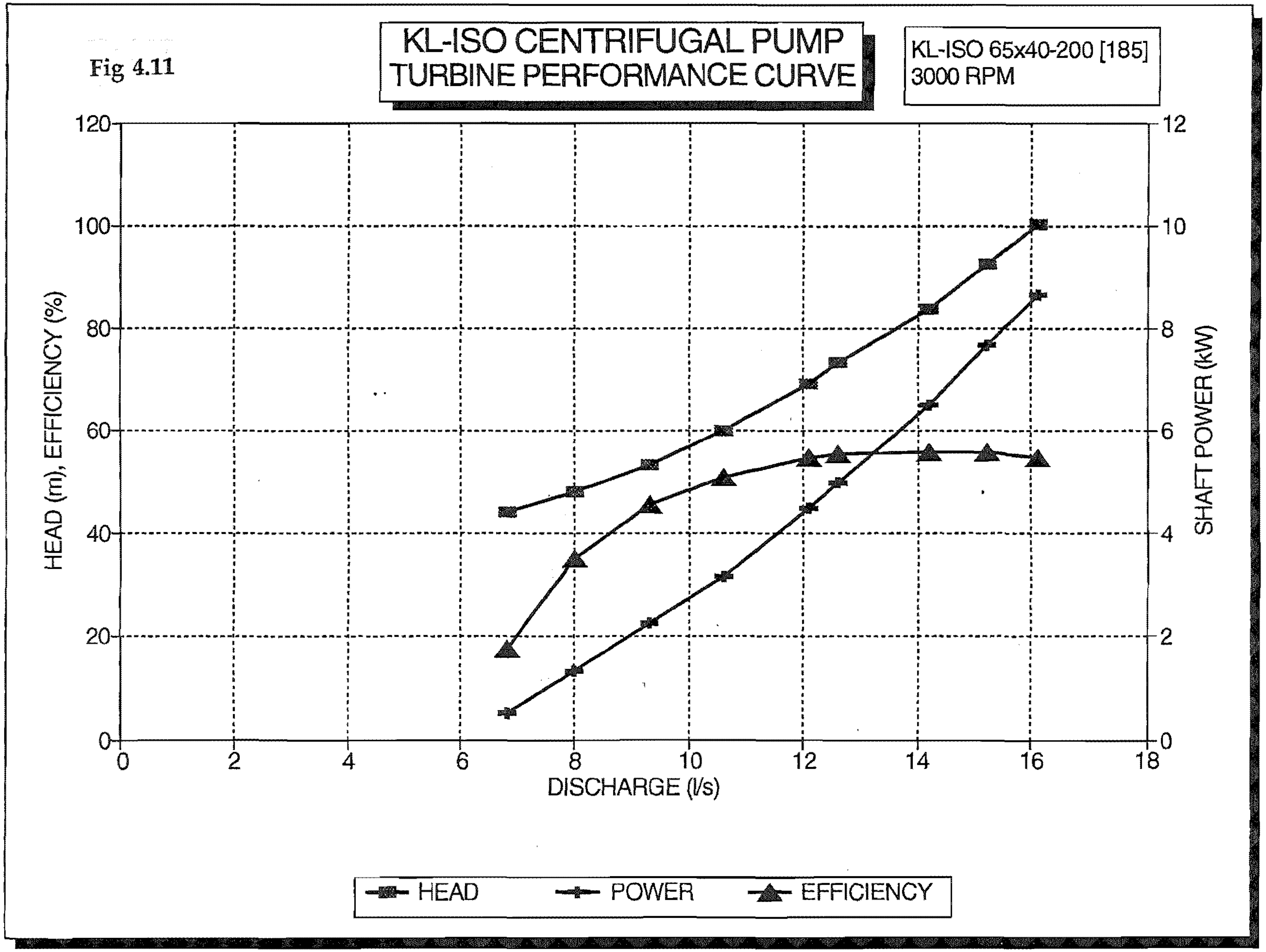

KL-ISO CENTRIFUGAL PUMP
TURBINE PERFORMANCE CURVE
KL-ISO 65x40-200 [185]
3000 RPM
HEAD (m), EFFICIENCY (%)
SHAFT POWER (kW)
DISCHARGE (l/s)
0
20
40
60
80
100
120
0
2
4
6
8
10
12
0
2
4
6
8
10
12
14
16
18
HEAD
POWER
EFFICIENCY

Fig 4.12

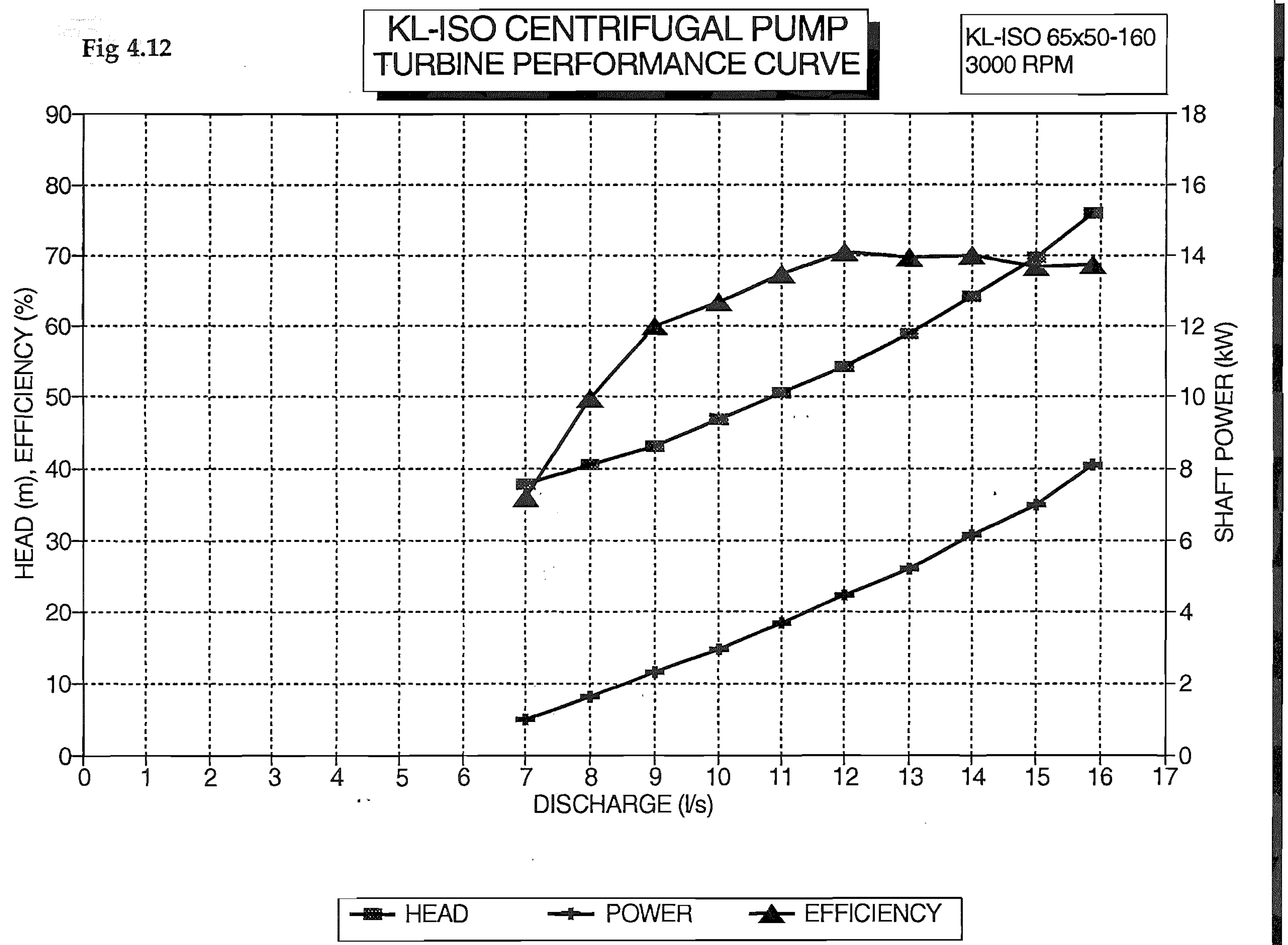
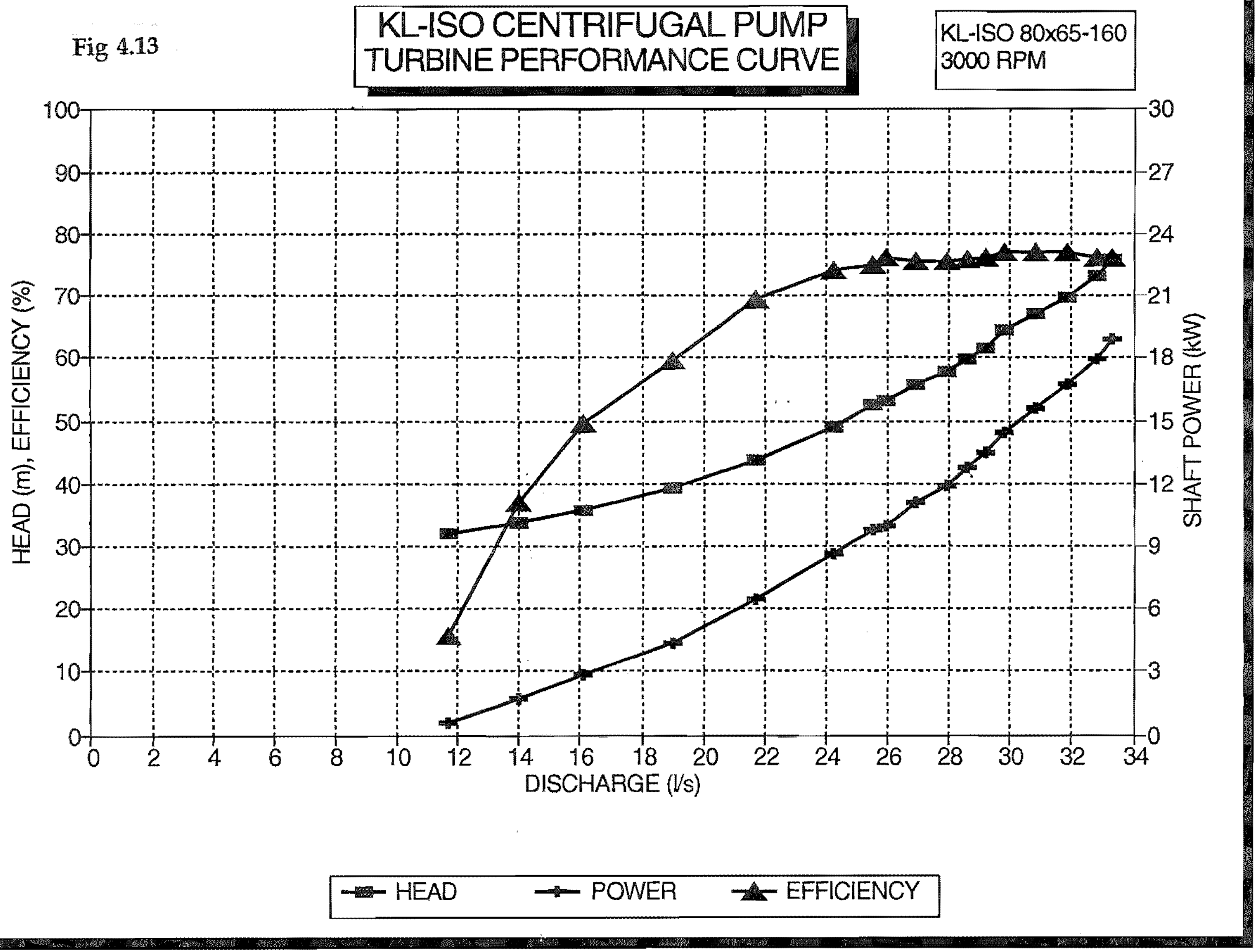
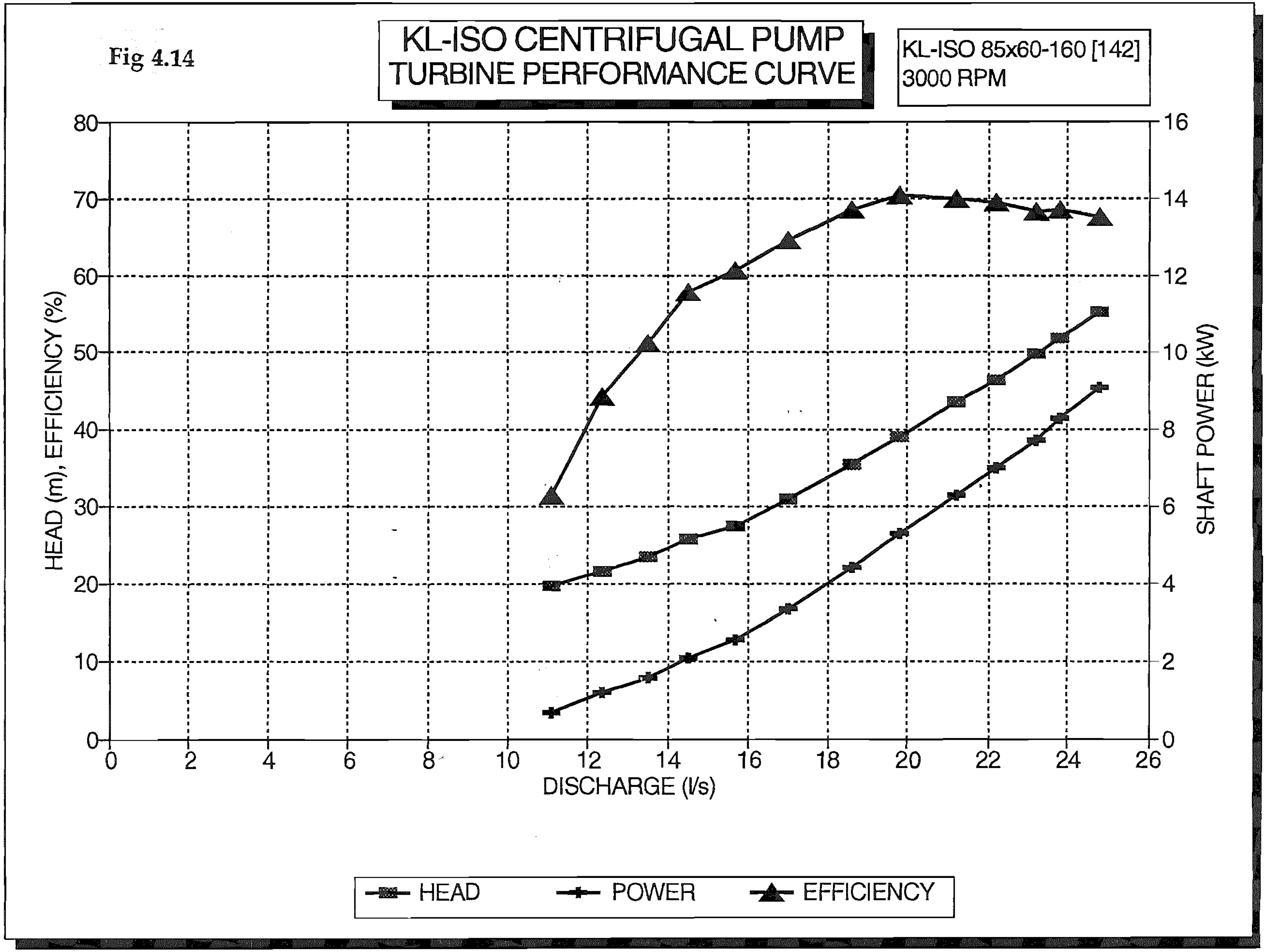

KL-ISO CENTRIFUGAL PUMP
TURBINE PERFORMANCE CURVE
KL-ISO 65x50-160
3000 RPM
HEAD (m), EFFICIENCY (%)
SHAFT POWER (kW)
DISCHARGE (l/s)
0
10
20
30
40
50
60
70
80
90
0
2
4
6
8
10
12
14
16
18
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17
HEAD
POWER
EFFICIENCY

Fig 4.13



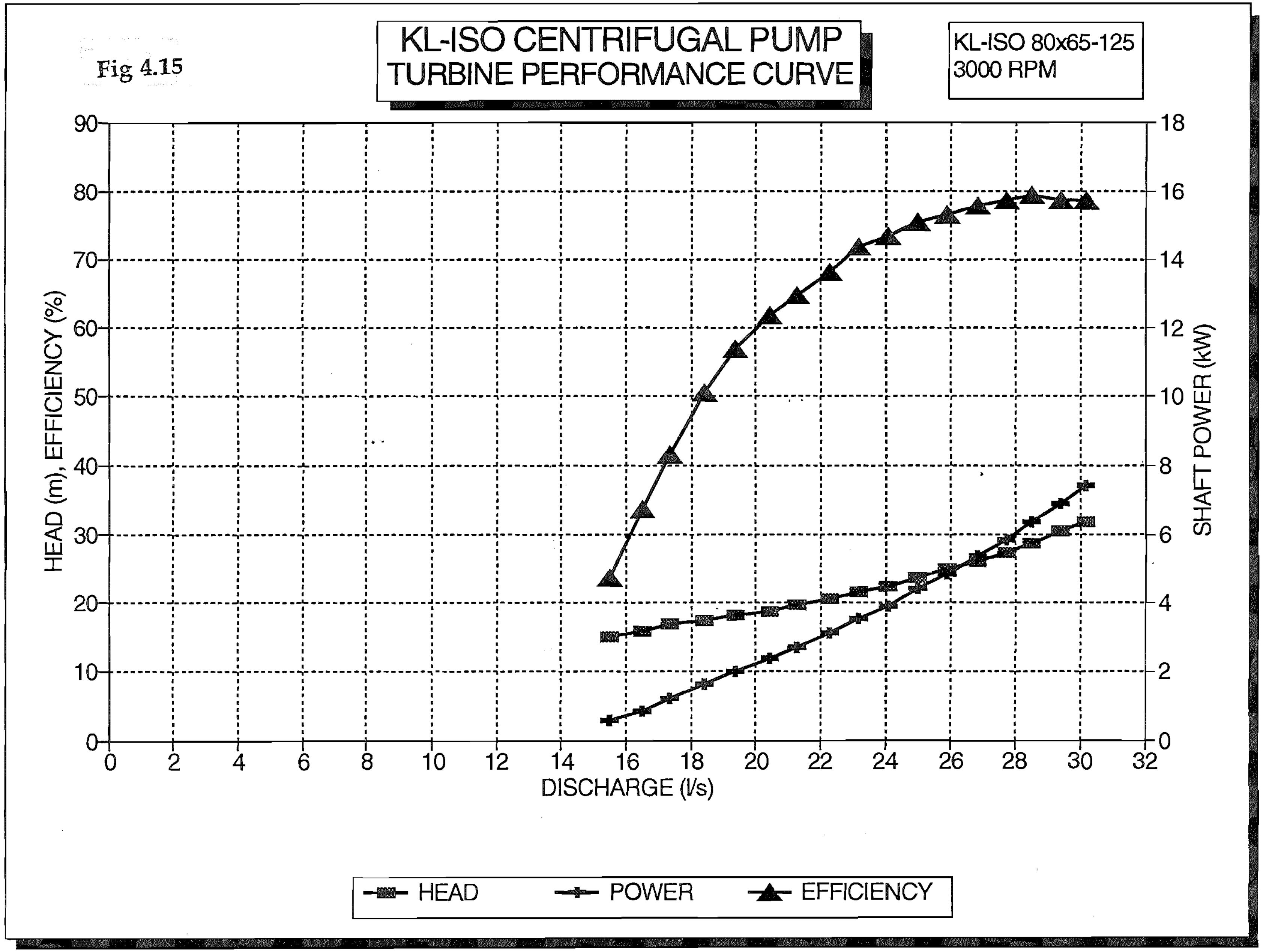
KL-ISO CENTRIFUGAL PUMP
TURBINE PERFORMANCE CURVE
KL-ISO 80x65-160
3000 RPM
HEAD (m), EFFICIENCY (%)
0
10
20
30
40
50
60
70
80
90
100
SHAFT POWER (kW)
0
3
6
9
12
15
18
21
24
27
30
DISCHARGE (l/s)
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
32
34
HEAD
POWER
EFFICIENCY

Fig 4.14



KL-ISO CENTRIFUGAL PUMP
TURBINE PERFORMANCE CURVE
KL-ISO 85x60-160 [142]
3000 RPM
HEAD (m), EFFICIENCY (%)
SHAFT POWER (kW)
DISCHARGE (l/s)
0
10
20
30
40
50
60
70
80
0
2
4
6
8
10
12
14
16
0
2
4
6
8
10
12
14
16
18
20
22
24
26
HEAD
POWER
EFFICIENCY

Fig 4.15



KL-ISO CENTRIFUGAL PUMP
TURBINE PERFORMANCE CURVE
KL-ISO 80x65-125
3000 RPM
HEAD (m), EFFICIENCY (%)
SHAFT POWER (kW)
DISCHARGE (l/s)
0 10 20 30 40 50 60 70 80 90
0 2 4 6 8 10 12 14 16 18
0 2 4 6 8 10 12 14 16 18 20 22 24 26 28 30 32
HEAD
POWER
EFFICIENCY

Fig 4.16

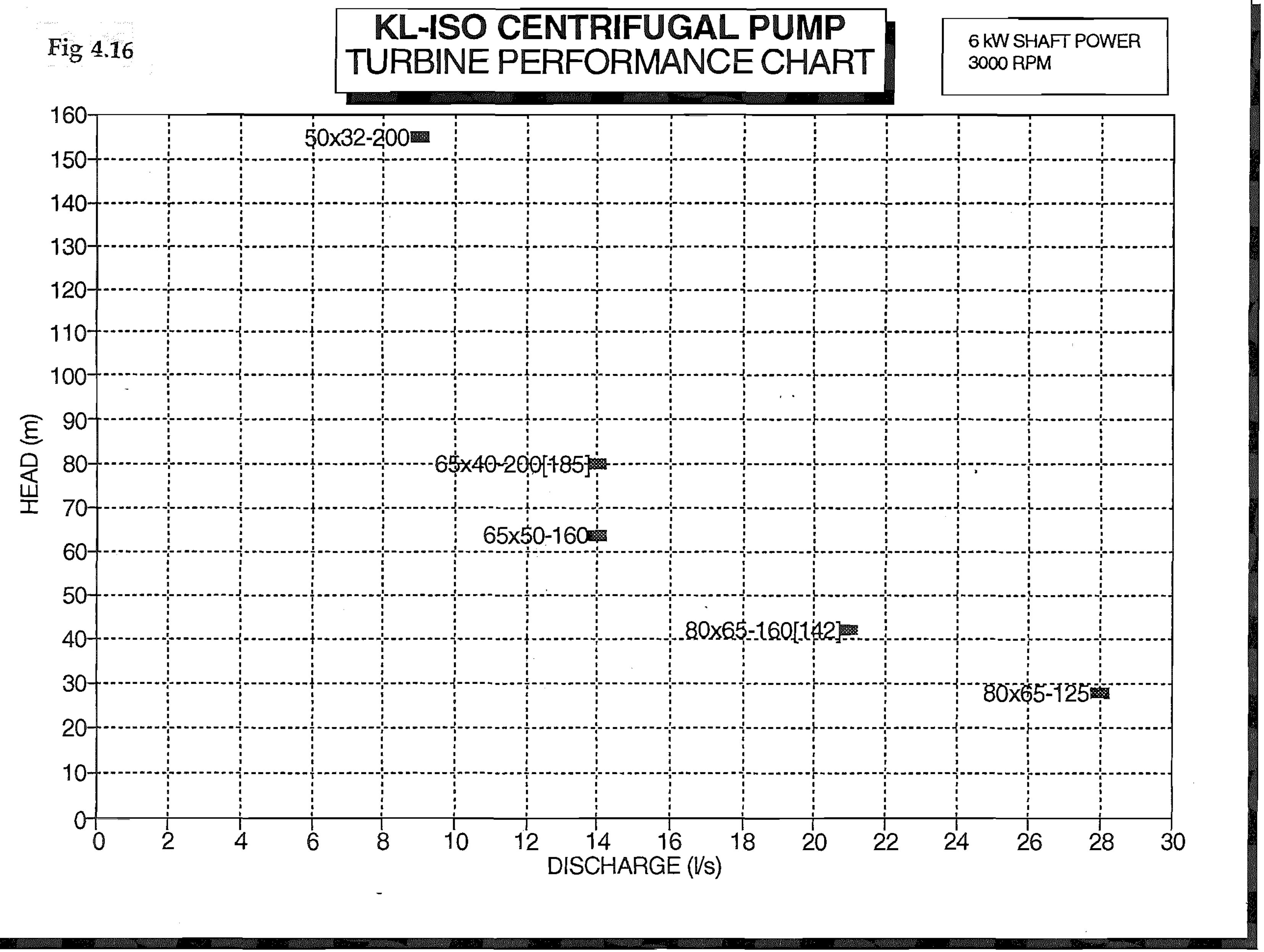
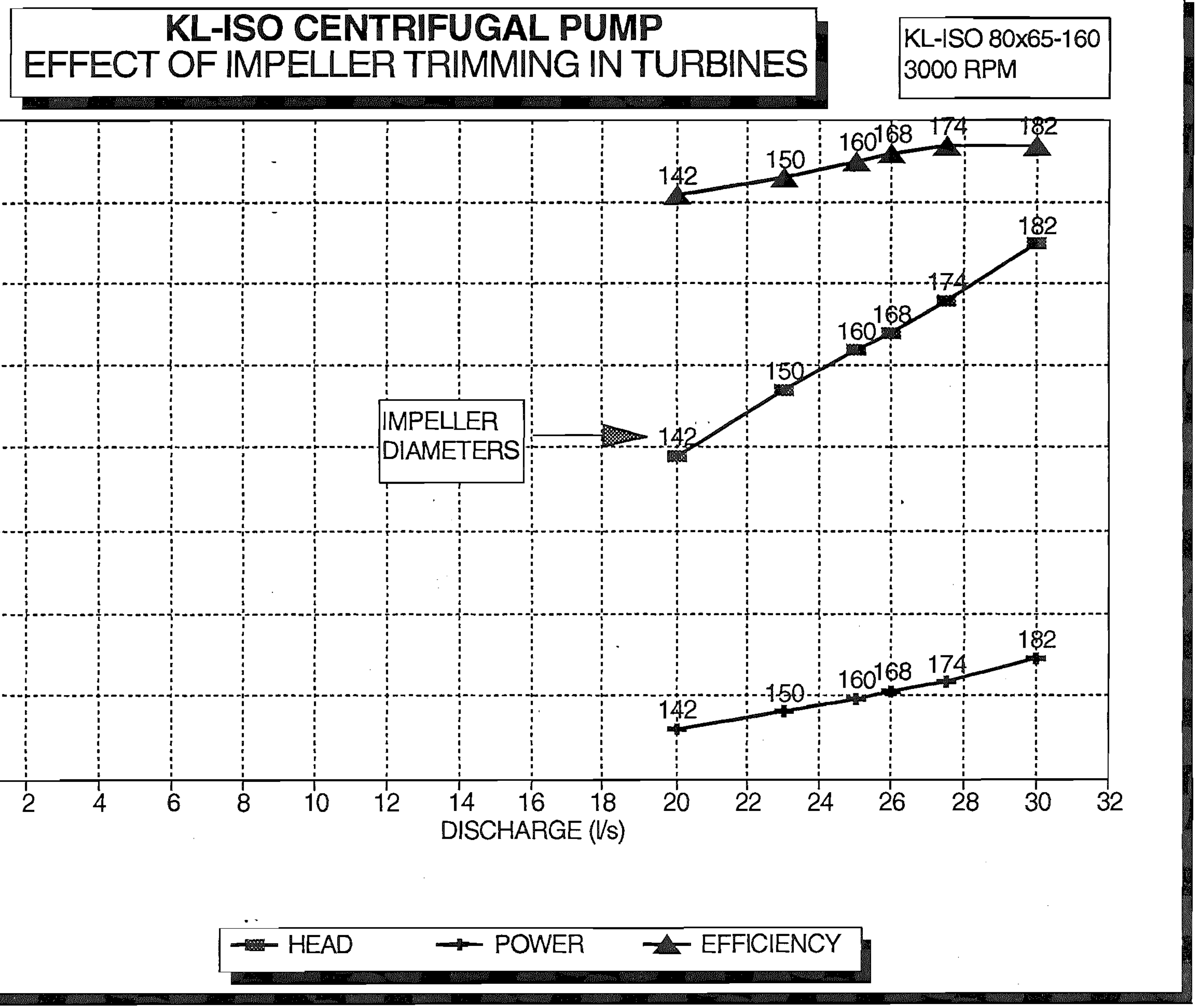

KL-ISO CENTRIFUGAL PUMP
TURBINE PERFORMANCE CHART
6 kW SHAFT POWER
3000 RPM
HEAD (m)
DISCHARGE (l/s)
50x32-200
65x40-200[185]
65x50-160
80x65-160[142]
80x65-125
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
160
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30

Fig 4.17



KL-ISO CENTRIFUGAL PUMP
EFFECT OF IMPELLER TRIMMING IN TURBINES
KL-ISO 80x65-160
3000 RPM
HEAD (m), POWER (kW), EFFICIENCY (%)
DISCHARGE (l/s)
IMPELLER DIAMETERS
142
150
160
168
174
182
0
10
20
30
40
50
60
70
80
0
2
4
6
8
10
12
14
16
18
20
22
24
26
28
30
32
HEAD
POWER
EFFICIENCY

Fig 4.18

KL-ISO CENTRIFUGAL PUMP
PUMP TO TURBINE CONVERSION CHART

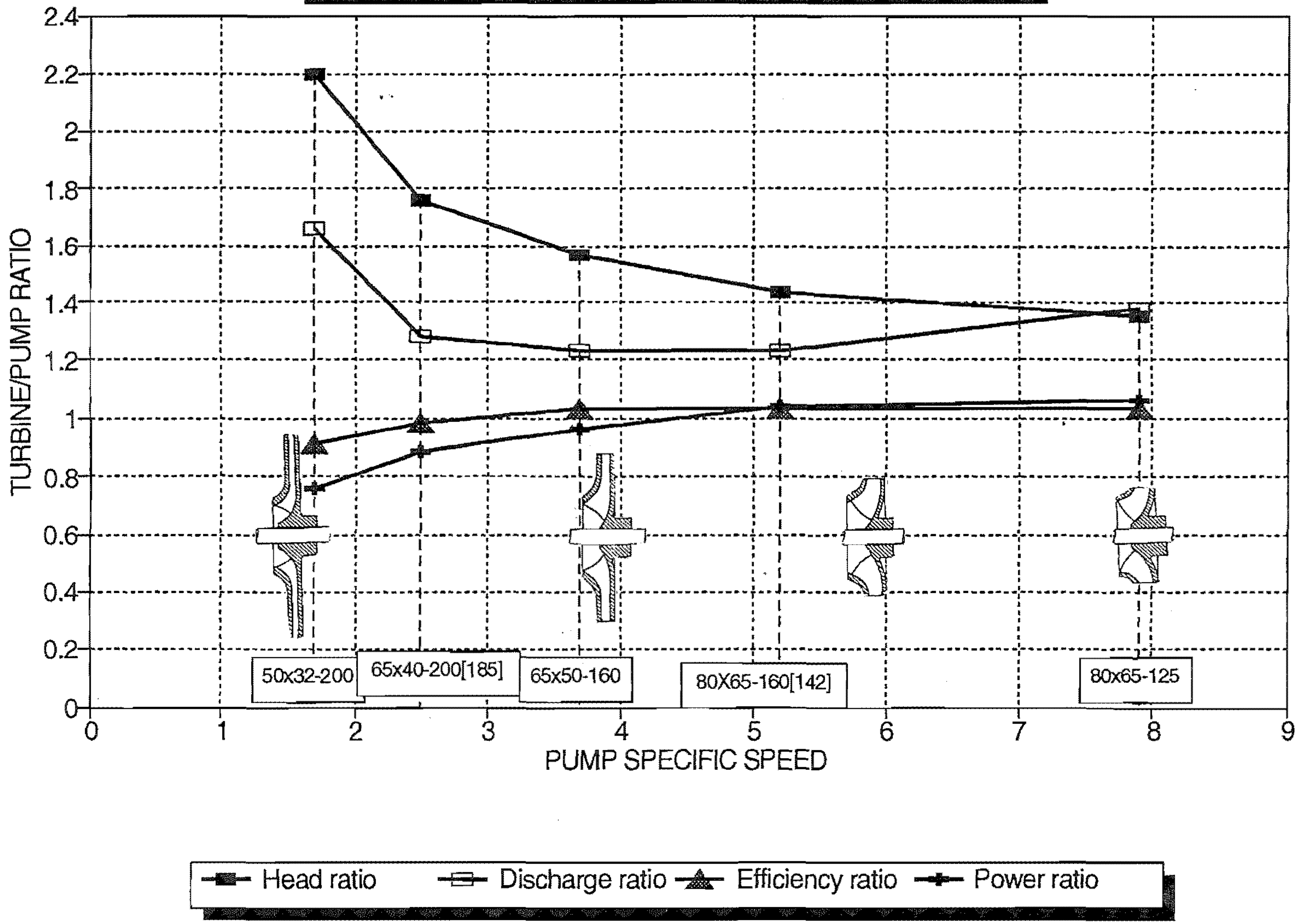
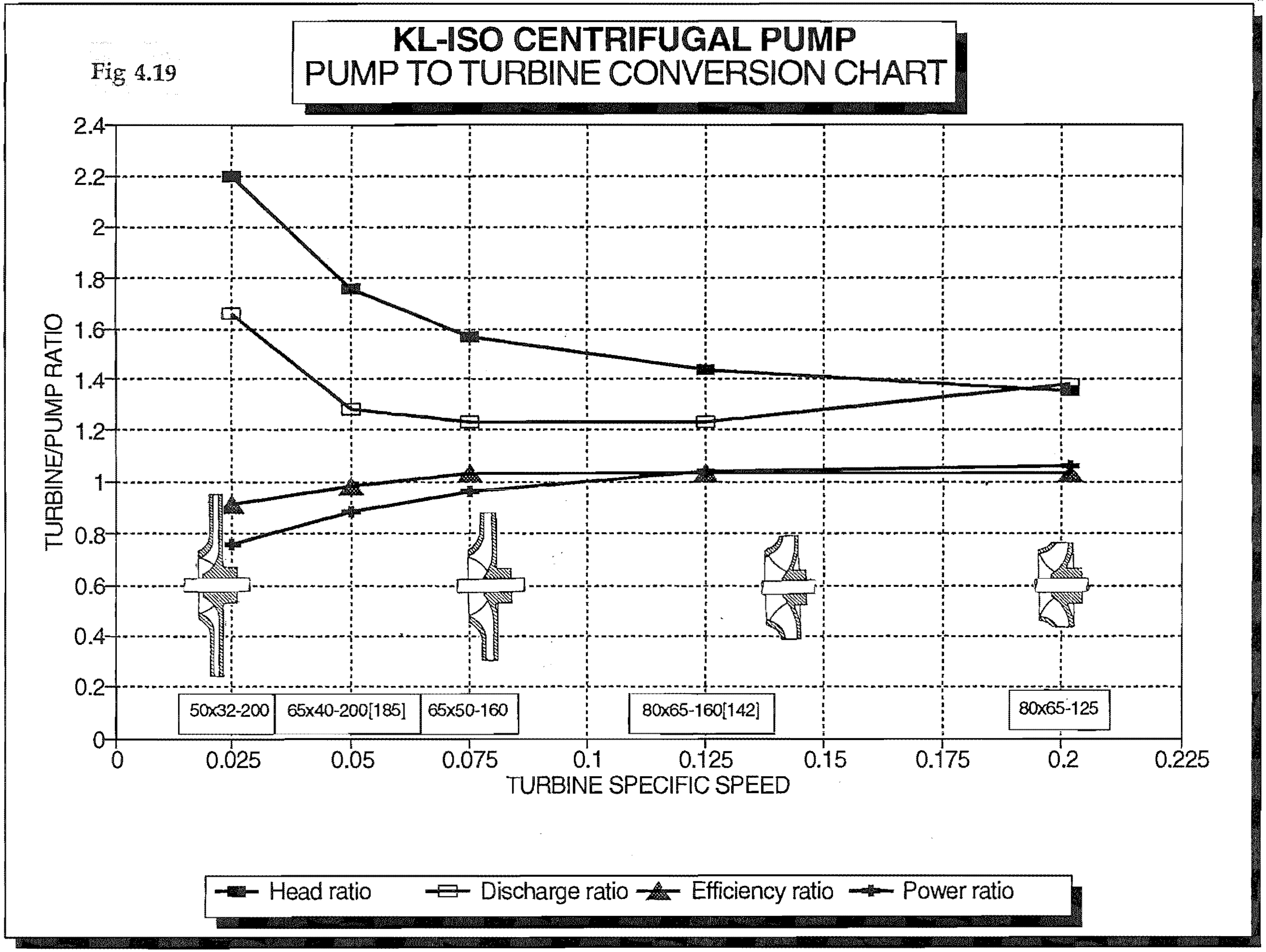


Fig 4.19


KL-ISO CENTRIFUGAL PUMP
PUMP TO TURBINE CONVERSION CHART
TURBINE/PUMP RATIO
2.4
2.2
2
1.8
1.6
1.4
1.2
1
0.8
0.6
0.4
0.2
0
0
0.025
0.05
0.075
0.1
0.125
0.15
0.175
0.2
0.225
TURBINE SPECIFIC SPEED
50x32-200
65x40-200[185]
65x50-160
80x65-160[142]
80x65-125
Head ratio
Discharge ratio
Efficiency ratio
Power ratio

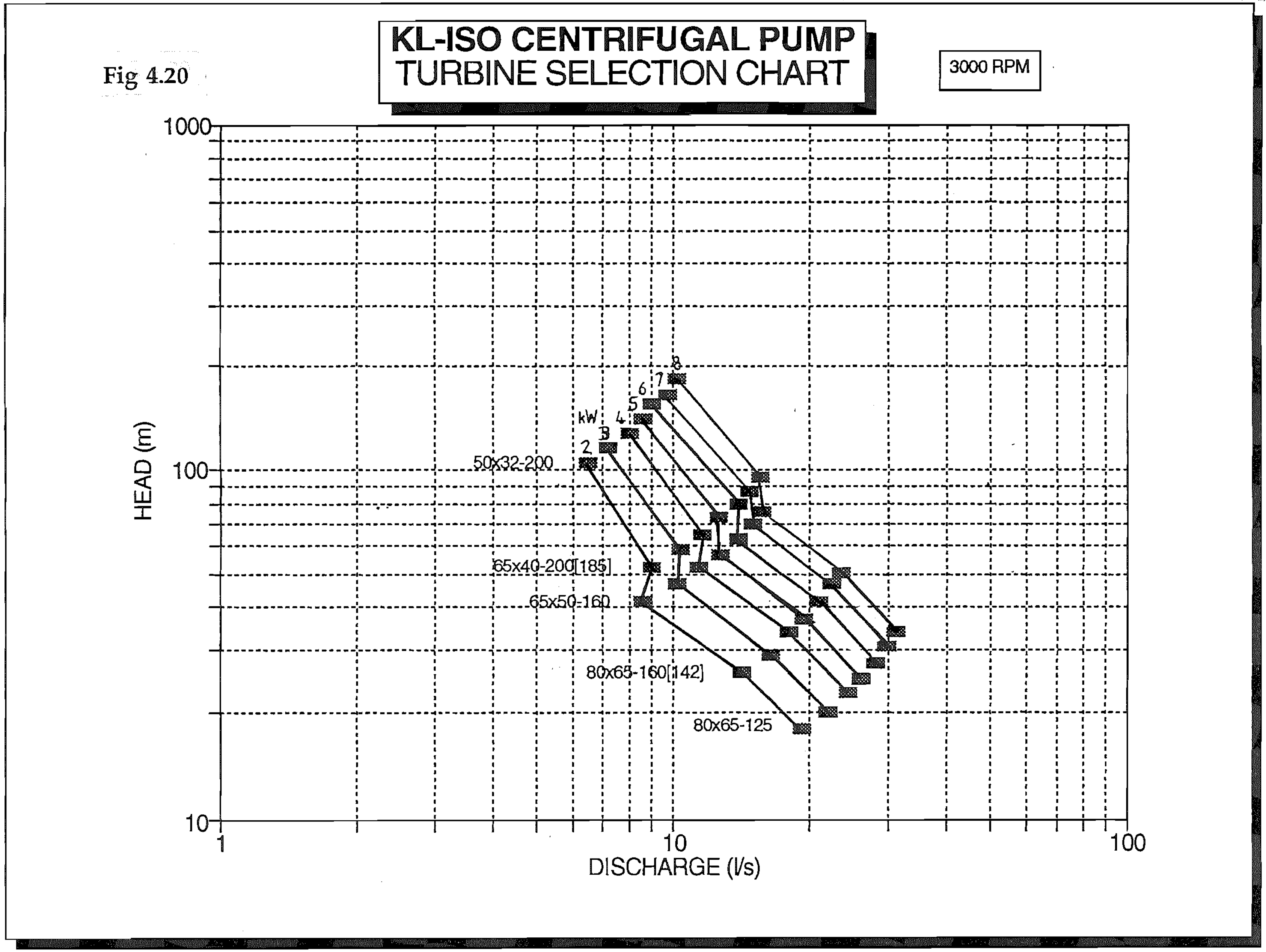
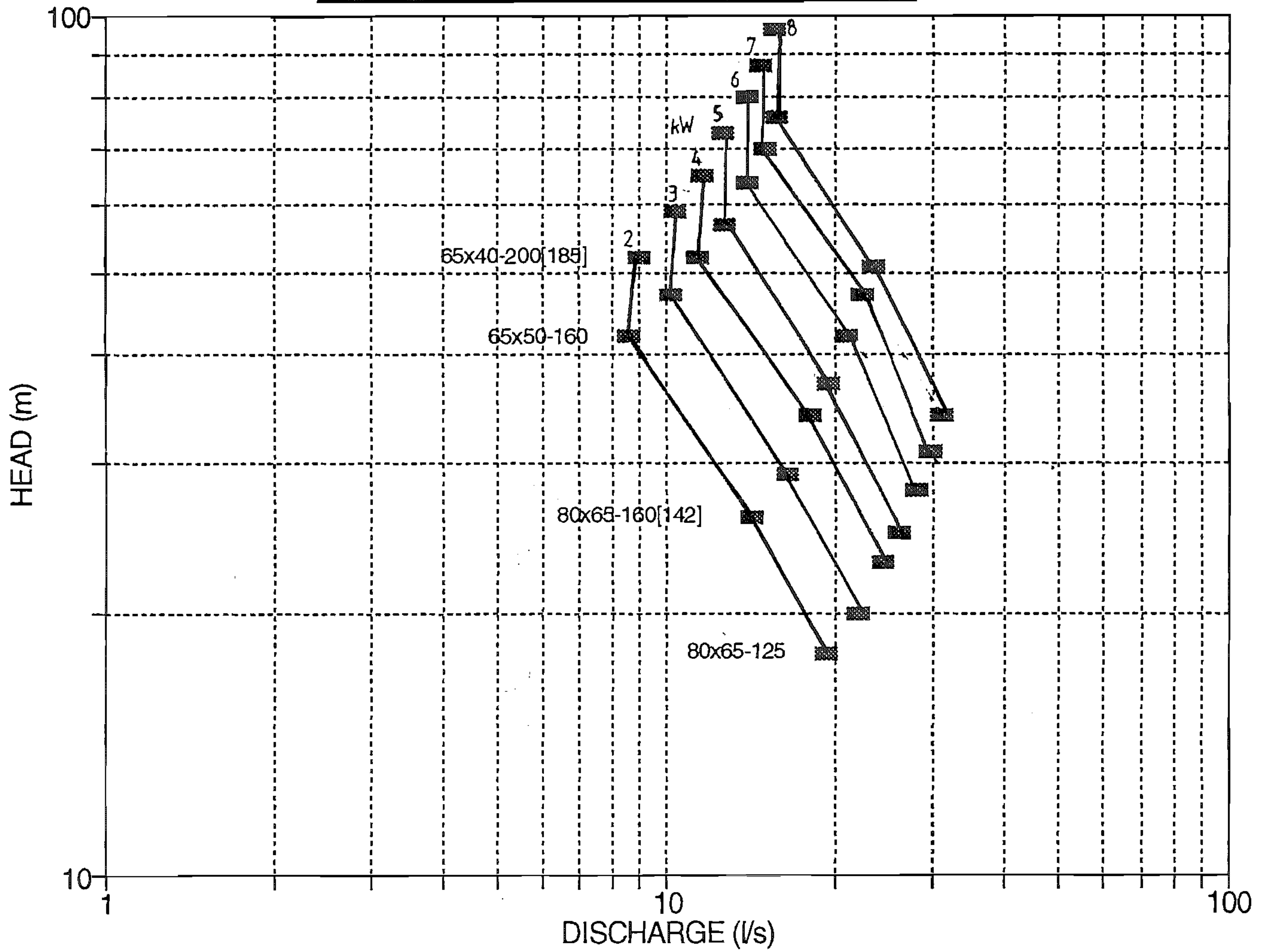

KL-ISO CENTRIFUGAL PUMP
TURBINE SELECTION CHART
3000 RPM
HEAD (m)
DISCHARGE (l/s)
1000
100
10
1
10
100
kW
2
3
4
5
6
7
8
50x32-200
65x40-200[185]
65x50-160
80x65-160[142]
80x65-125

Fig 4.20

Fig 4.21

KL-ISO CENTRIFUGAL PUMP
TURBINE SELECTION CHART

3000 RPM



CHAPTER FIVE

5 SELECTION OF GENERATOR

5.1 ELECTRICAL MACHINES

Electrical machines comprise three main types:-

1 Direct-current machines
2 Synchronous machines
3 Asynchronous or induction machines

Electrical machines that convert electricity to mechanical shaft power are called motors, while those that produce electricity from mechanical shaft power are called generators.

In stand-alone power generating applications, synchronous generators are the common type of electric machine used. Induction generators (asynchronous generators) on the other hand, have limited applications due to the inherent problems with excitation and voltage regulation (see section 5.3). However, researchers are devising ways that enable induction generators to operate in a stand-alone manner [2,41]. Direct-current machines are normally not used in this application.

This chapter explores the operation of both synchronous and asynchronous generators and outlines their advantages and disadvantages.

5.2 SYNCHRONOUS GENERATORS

Synchronous machines are so called because they operate without slip. The two basic parts of a synchronous machine are the magnetic field structure, carrying a dc-excited winding, and the armature. The armature is stationary while the magnetic field structure rotates. The dc winding on the rotating field structure is connected to an external source through slip rings and brushes. In a "brushless" exciter, the dc current is obtained from a separate ac winding placed on the main rotor, or in some cases on a separate rotor, directly connected to the main rotor. The ac voltage is rectified in a rectifier circuit placed on the rotor.

Depending on rotor construction a synchronous machine may be a round-type (Fig 5.1) or salient-pole type (Fig 5.2).

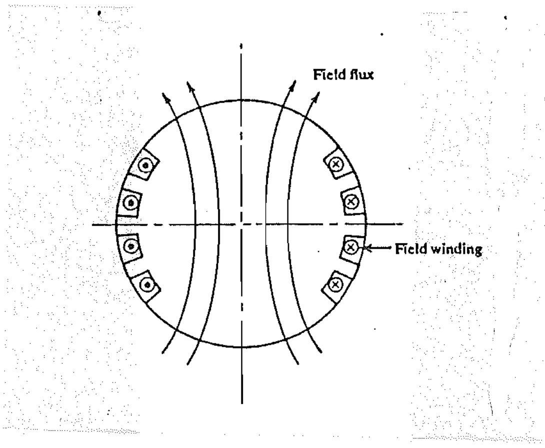


Fig 5.1 Field winding on a round rotor

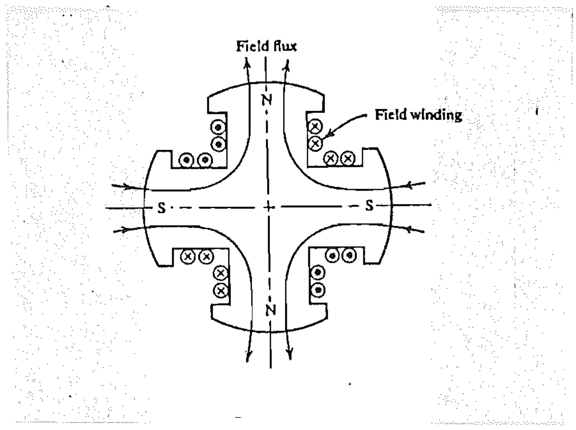


Fig 5.2 Field winding on a salient rotor

a) Torque/speed Characteristics and Synchronous Speed

A synchronous machine can only run at speed ,

$$N_s = \frac{120f}{p}$$

Where, p is the number of pairs of poles. i.e. for a 2-pole synchronous machine in a 50 Hz system, the speed would be 3000 rpm. This is known as synchronous speed.

The torque/speed characteristic of synchronous generators is shown in Fig 5.3. The vertical line indicates that the torque can vary up to the maximum value T_{max} at synchronous speed. It may also run as a synchronous motor. Between minimum and maximum torque the machine maintains synchronism by a variation in phase angle. If the turbine torque rises above T_{max}, the generator cannot absorb all the turbine power, which may lead to runaway speed, and the system is said to be out of synchronism. Synchronous generators are available from 1 kW for portable generators up to 1 MW for large power generating plants.

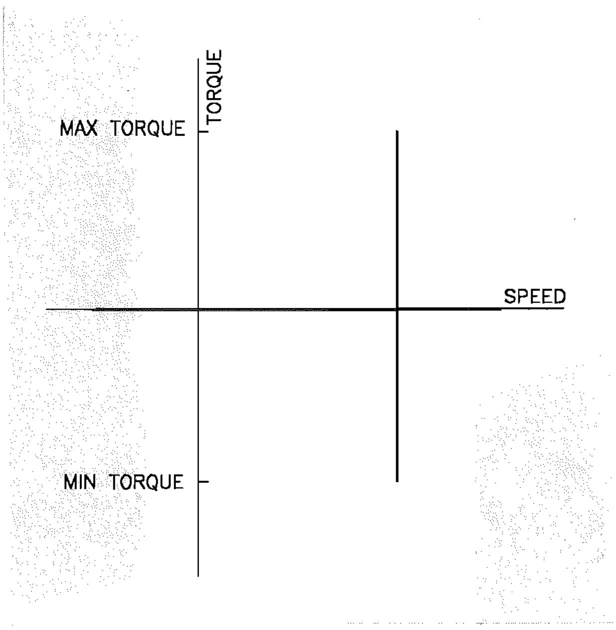


Fig 5.3 Torque/speed characteristic of a synchronous machine.

5.3 INDUCTION GENERATORS

An induction generator, on the other hand, requires some form of external excitation to operate. Most often, this is achieved by connecting the generator to an energised electrical network, but this is not possible for an isolated power plant. Induction generators are essentially induction motors driven as generators at speeds greater than synchronous, i.e. with negative slip. They are simpler in construction and less costly than synchronous generators. In addition they are more rugged in design and therefore well suited for conditions encountered by most microhydro applications. For these reasons, many efforts have been made to design electronic devices for the excitation and control of induction generators at isolated sites [2]. However, such devices have only begun to find their way in to the marketplace.

a) Torque/Speed characteristics of an induction generator

Torque/speed characteristic of an induction generator is shown in Fig 5.4.

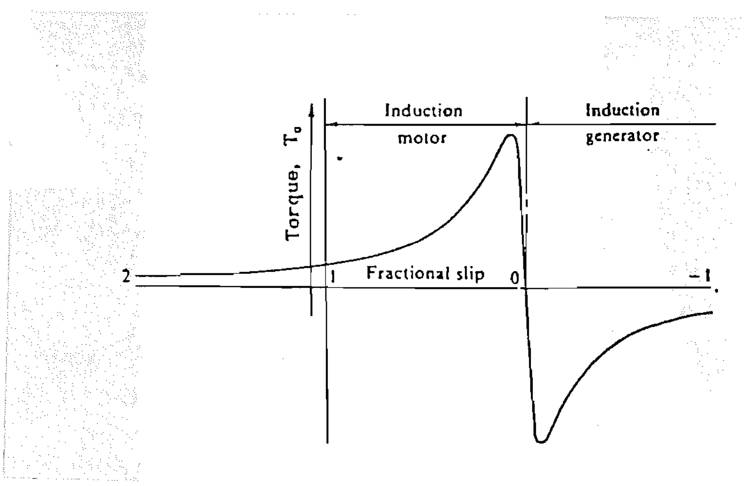


Fig 5.4 Torque/speed characteristic of an induction generator

5.4 SYNCHRONOUS VS INDUCTION GENERATORS

There are advantages and disadvantages in using the different types of generators which are outlined below:-

INDUCTION GENERATOR	SYNCHRONOUS GENERATOR
1 Rotor construction	
Uninsulated copper bars	Insulated wire or strap
Relatively few conductors	Many series turns
Firmly held in separate slots	Salient pole and round-type
Few connections	Many small connections
Few basic parts	Many basic parts
2 Excitation	
Require separate excitation system	Attached DC exciter
No brush or collector rings	Brushed or brushless
3 Generated waveform	
Tend to damp out harmonics	Tends to initiate harmonics
Passive element	Active element in system
	Controls voltage and frequency
4 Costs	
Lower first costs	Higher first costs
Low maintenance	Regular maintenance
Lower efficiency	High efficiency
Lagging power factor	Leading power factor possible

5.5 GENERATOR FOR THE MICROHYDRO GENERATING SET

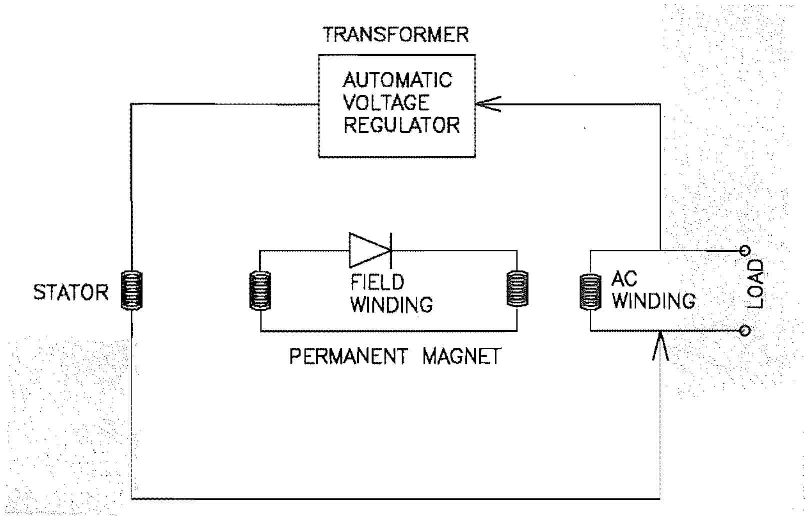


Fig 5.5 Schematic layout of a brushless synchronous generator

Synchronous generators are most easily utilised in stand-alone microhydro applications. However, they do have a number of disadvantages. Small alternators (generators) usually have a high transient impedance which results in poor starting of motors and large transient voltage droops. In addition, unbalanced loading will result in significant harmonic distortions in the voltage generated. Harmonic distortion of the synchronous generator can be reduced to an acceptable level by providing inertia on the rotating shaft to limit the fluctuations in speed due to unbalanced loading. The amount of inertia required on the shaft for the 5 kW microhydro generating set will be outlined in Chapter 6.

The generator to be used in the microhydro installation will be a synchronous alternating current generator of brushless type with self induced excitation and automatic voltage regulation. These features are required for stand alone applications as outlined earlier. A Markon B21D, single phase, 2 pole, AC, brushless synchronous generator with self induced excitation system and automatic voltage regulation is specified. The generator can produce up to 6 kW at unity power factor or 4.8 kW at 0.8 pf.

The Markon generator has a conventional wound stator and a salient pole retaining field system, incorporating on a common shaft, the armature of the ac exciter and a full-wave diode main field rectifier. The machine is self-excited from the main stator winding though an automatic voltage regulation (AVR).

a) Generator operations

It is important that the generator must meet the load demand. In this case, the generator is sized at 6 kVA. Similarly, the load demand must not exceed 4.8 kW at 0.8 pf or the generator will overload.

b) Power Factor

$$Power\ Factor = \frac{kW}{kVA} \tag{29}$$

Power factor is a very important parameter when considering electrical power applications such as microhydro generation. It is defined as the ratio of useful electrical power to apparent power. (Apparent power is the product of volts and amps). Therefore, power factor is determined by the connected load. In inductive circuits for example, current lags voltage. Fig 5.6 represents the current lagging corresponding voltage by 37 deg. This represents a power factor of 0.8. Where both are positive, or both negative, the resulting power is positive. This is represented by the shaded area above the zero line. When current or voltage are of opposite value, the resulting power is negative and is represented by shaded areas below the zero line.

Net power	- Positive area minus negative area
Apparent power	- Sum of the two areas
Power factor	- Net power divided by apparent power

Mathematically, power factor equals the cosine of the angle by which current lags, or in rare cases, leads the voltage.

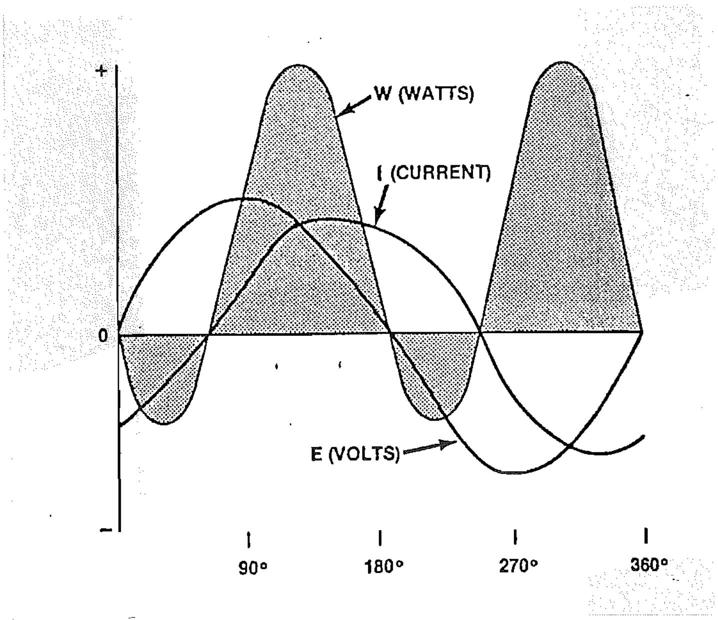


Fig 5.6 Electrical power characteristics

Load	Power factor	Phase shift
Resistive load -electric heaters -incandescent lights	1	Voltage and current in-phase
Inductive load -Electric motors -Fluorescent lights	< 1	Current lags voltage
Capacitive load -Uncommon	> 1	Current leads voltage

c) Generator testing

Tests on the Markon generator were conducted using an experimental rig. The results are outlined in Chapter 6. One of the prime objectives was to determine the generator's compatibility with the electronic load governor.

CHAPTER SIX

6 LOAD GOVERNING

6.1 THE NEED FOR LOAD GOVERNING

The electronic load governor is the single most important item making possible cost reductions in microhydro installations. Not only does it replace the costly conventional mechanical speed governor, but it also allows constant flow operation of turbines and thus eliminates any problem from hydraulic instability. In addition it enables virtually any turbine to be used including centrifugal pumps in the reverse mode of operation.

The function of the electronic load governor is to control the speed and frequency of small isolated hydro electric plants by automatic control of the total electrical demand presented to the AC generator, that is, to divert automatically the unused electrical load to banks of dummy resistive load. The dummy loads may be resistors in air or water [11].

The electronic load governor which was initially manufactured by Delphi Industries Ltd of Auckland has now been modified and is presently manufactured by Diesel and Powers Systems Ltd of Dunedin. During the development of the 5 kW microhydro, the new version of the governor was released by Diesel and Powers and has slightly different features from the original governor. Little information on the new version is available. However, it is understood that the principle of operation remains unchanged. (Data sheets for both the versions of the governor are shown in appendix 6). This chapter investigates the operation and the behaviour of the Delphi electronic load governor in conjunction with the synchronous generator selected in Chapter 5.

6.2 OPERATION OF THE ELECTRONIC LOAD GOVERNOR

The governor operates by varying the control ballast load in response to variations in the generated frequency. If the consumer load is increased, a decrease in turbine speed will occur with a consequent drop in electrical frequency causing the load governor to reduce the ballast load accordingly.

Like any conventional speed governor, the electronic load governor has a 'frequency droop' being the percentage change in frequency as the load governor operates over its full control range. The Delphi governor has a fixed frequency droop of approximately 3%, but the new version governor has an adjustable droop of 2 or 4%.

The governor may be used for single or three phase application. For three phase, three governors are required, i.e. one for each phase. Only single phase operation is outlined in this study. The operation of the governor is illustrated by Fig 6.1. Actual configuration of the Delphi electronic load governor is shown in Fig 6.9B.

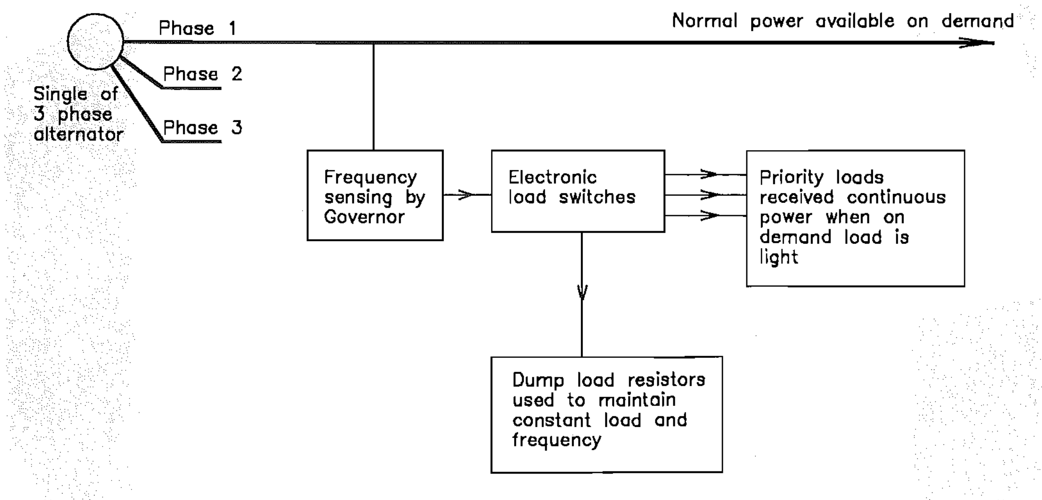


Fig 6.1 Operation of the electronic load governor

The governor uses zero voltage switching where the moment of switching corresponds to voltage zero. This minimises harmonic distortion and radio frequency interference (RFI). The switching between different loads implies that the control is not stepless. The number of steps and their relative sizes is designed to be 15. The load switching characteristic is shown in Fig 6.2.

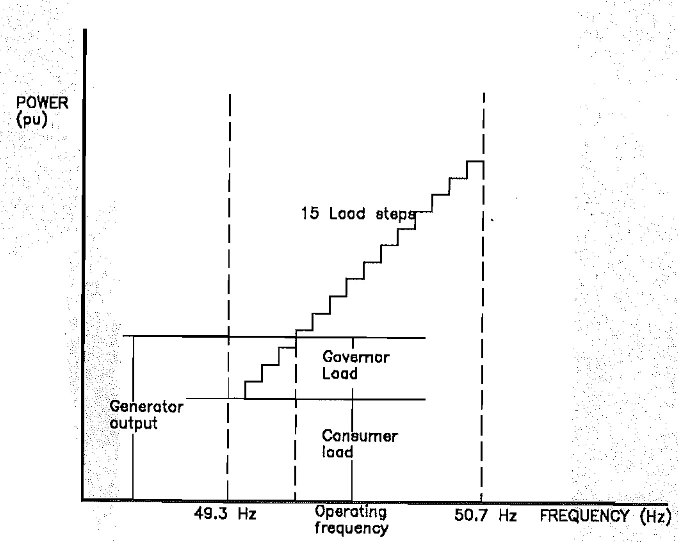


Fig 6.2 Power frequency characteristics of the electronic load governor

The interval between successive negative-going zero-crossings of the phase voltage waveform is measured by counting pulses from a crystal controlled reference clock set at 26 kHz for a 50 Hz system. Surplus counts above a threshold of 512 are fed to a 4 bit counter so arranged that 15 counts corresponding to 49.3 Hz will result in binary outputs 0000, while zero counts corresponding to 50.8 Hz, result in binary outputs 1111. At the next positive-going zero-crossing of the voltage, the binary output of the 4 bit counter is used to switch 4 triacs controlling 4 dump loads, sized in the ratio 1:2:4:8 as in Fig 6.3.

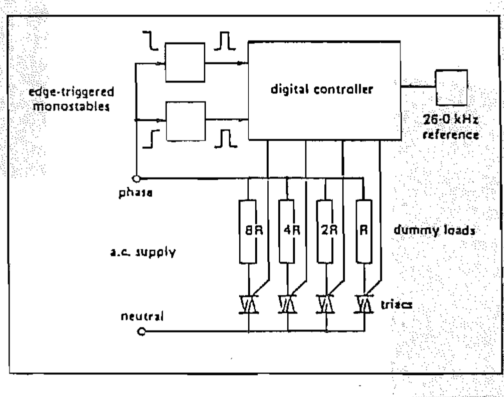


Fig 6.3 The Delphi electronic load governor hardware

The governor described above, controlled four loads in the ratios 1:2:4:8 thereby being able to provide any load between 1 and 15 times the smallest load, in equal steps by appropriate combinations of the four loads. As the governor applies more loads (i.e. as the excess power generated increases) it switches amongst the four loads. If a load of 3/15 per unit (pu) is required, the loads corresponding to 1/15 pu and 2/15 pu would be applied, and an increase in power absorbed would result in these two loads being removed and the load corresponding to 4/15 pu applied [59].

In operation, an input frequency of 49.3 Hz or less corresponds to a count of zero and all triacs will be switched off. At exactly 50 Hz, a load of 7/15 pu will be switched on. If the frequency is above 50.7 Hz, all triacs will be switched on. The switched load is proportional to frequency over the active range.

Because the governor is sensitive to frequency, which is the same throughout the supply system, governors may be installed at any point desired throughout that system. Fig 6.4 and Fig 6.5 show how the load governor may be employed in practice in a typical single and three phase application respectively.

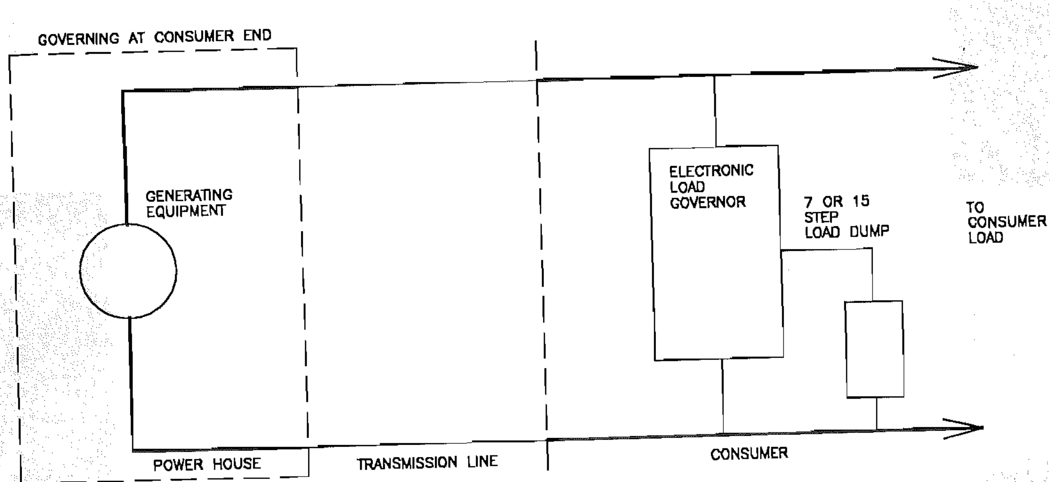


Fig 6.4 A typical single phase application

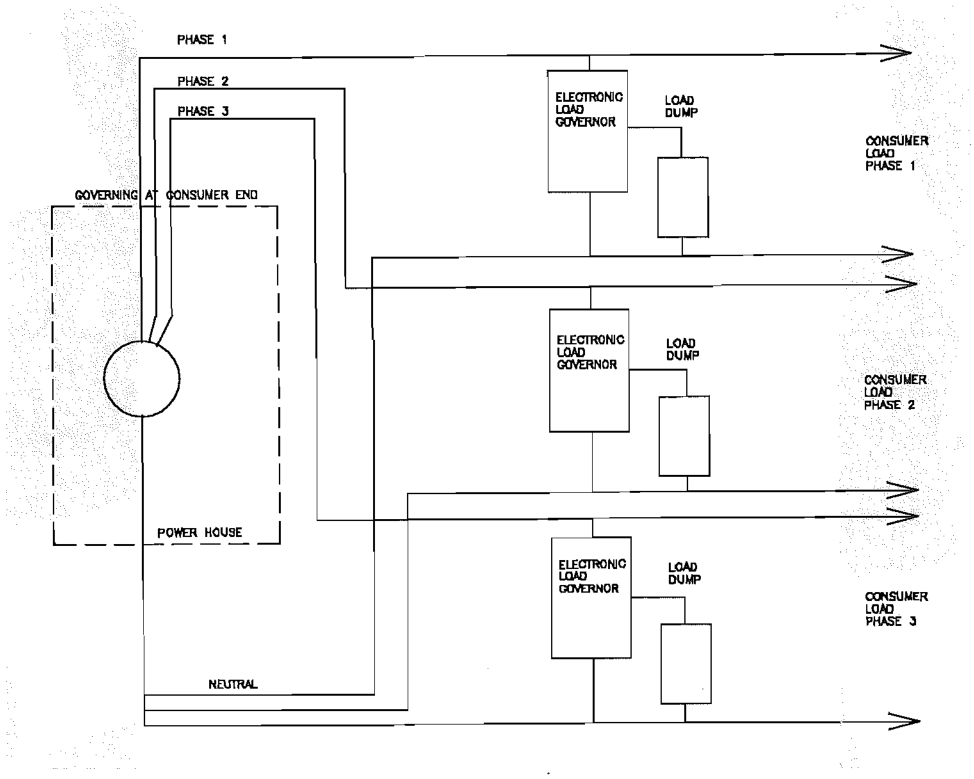


Fig 6.5 A typical 3 phase application

6.3 BALLAST LOADS

The energy dumped into the governor load may be discharged to waste or used to provide domestic hot water or space heating. The application where part of the dump load is to be utilised, requires the use of the governor in a mode which separates the controlled load into three dump loads in the ratios 1:2:4 (operated as before) and three low priority loads. The low priority loads are the first of the controlled loads to be applied and they are switched in sequence as power becomes available. Thus, they will remain connected until the consumer (high priority) load approaches full rated load. This provides a guaranteed supply of power to these controlled loads when there is any excess power in the system. Fig 6.6 shows this type of application.

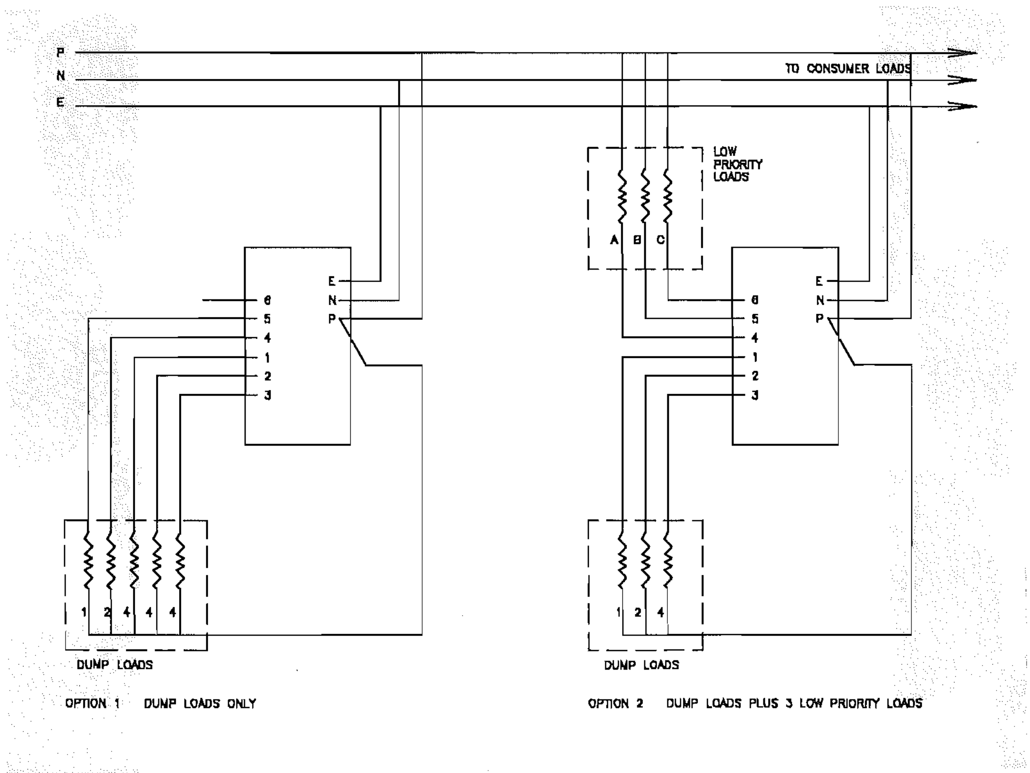


Fig 6.6 The electronic load governor ballast loads

6.4 TYPE OF BALLAST LOAD

Arbitrary switching means that the ballast loads have to be resistive. The ballast loads can be either resistors in air or resistors in water. Whichever one is chosen will be susceptible to failure. Failure of the ballast load will cause the speed to rise with the consequent effect of increasing the frequency. However, the microhydro generating set has a fail safe protection system as will be described in Chapter 7, which trips and stops the generator on over frequency or line voltage failure.

Failure of the ballast load can occur in the following ways:-

i) Elements in air

Elements in air can fail due to oxidation or the fire hazard due to radiant heat. It is therefore recommended that kiln wire elements housed in a furnace-type enclosure are used for this application, or if the dummy load is for space heating or any other similar application, ensure that it cannot set fire to the surroundings.

ii) Elements in water

Elements in water on the other hand can fail due to the elements not being fully immersed, vibration induced by the water current, or deposits of lime. As an added safety protection each dummy resistor could be fitted with a relay to release the brakes if the resistor failed.

6.5 USE OF MULTIPLE GOVERNORS IN SINGLE PHASE APPLICATION

Multiple governor units may be used in a single phase system where it is desired to control dump loads and low priority loads at two or more locations. For example, it may be desired to control water heating (dumped or low priority) and space heating (low priority) in 2 houses and a shed, and in addition provide for back-up load governing adjacent to the turbine. This is illustrated diagrammatically in Fig 6.7.

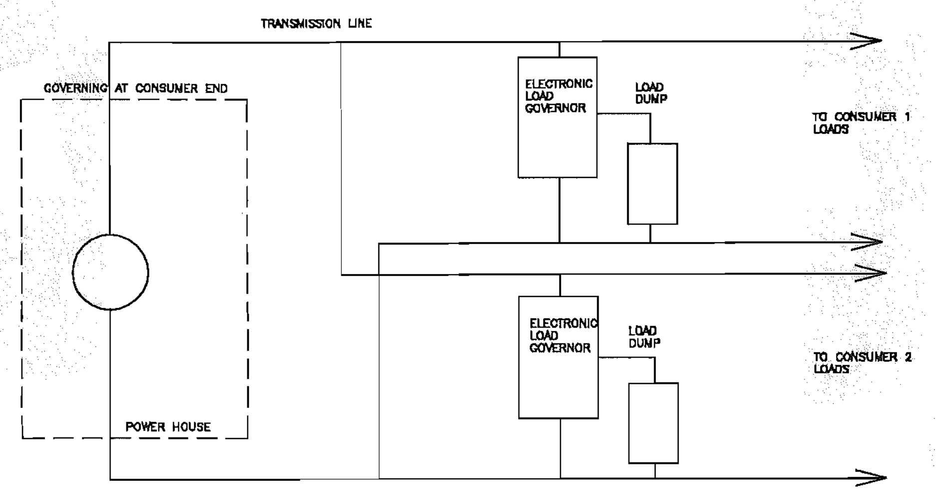


Fig 6.7 Possible use of multiple governors in single phase system

6.6 STABILITY OF THE ELECTRONIC LOAD GOVERNOR

The electronic load governor relies on obtaining an accurate measurement of the frequency from the period of the voltage waveform. However, for a synchronous generator, the transient reactance of the machine will cause a shift in phase of the output voltage whenever a change in external load occurs. As shown in Fig 6.8 for a load increase, the (sudden) change in phase of the terminal voltage results in an incorrect period measurement. The governor reduces the controlled load with the result that the phase changes in the opposite sense. This procedure can repeat with the governor both applying and removing the load alternately at each switching. If the amount of load removed is greater than the original load step then the system will become unstable at the switching frequency (25 Hz). A bandpass filter is used so that this instability does not arise. However, the filter introduces a delay into the response of the governor (to changes in frequency) with the result that the system may then become unstable at low frequencies (approximately 2 Hz). The proposed solution to this instability is the addition of inertia to the rotating shaft to limit the rate at which the speed can change.

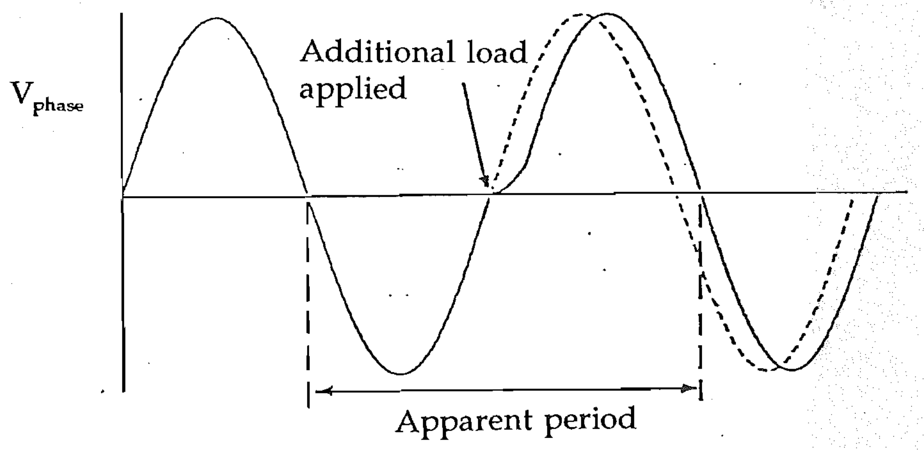


Fig 6.8 The switching of load causes a change in phase of the voltage and a resultant incorrect period

For an induction generator, any transient change in the applied load must be supplied via the machine's transient reactance which is typically lower than the transient reactance of the synchronous generator. Switched load changes applied to the induction machine, however, result in sudden frequency changes which may outweigh the phase changes found to be important for the synchronous generator. Thus the conditions for stability are quite different for synchronous and asynchronous machines.

Inertia in the rotating machine is therefore an essential requirement of a system using the Delphi load governor. Lack of inertia will result in instability. Inertia to facilitate system stability is ensured by the addition of a flywheel to the machine shaft. Delphi Industries have produced a set of empirical expressions which estimate the flywheel inertia required to produce a stable electrical output. These expressions are outlined as follows:-

$$I = (0.02)(Generator\ ratingkW)\left(\frac{3000}{RPM}\right)^2 \tag{E6.1}$$

For a disc flywheel of mass m and diameter d, then,

$$I = \frac{md^2}{8} \tag{E6.2}$$

For a steel disc, the thickness t is then,

$$t(mm) = \frac{0.18m}{d^2} \tag{E6.3}$$

For the 5 kW microhydro generating set, the generator is rated at 6 kW, 2 pole and operates at 3000 rpm.

I = 0.11 kgm^2
m = 8 kg
d = 330 mm (for t = 12 mm)

Experience with earlier microhydro installations have shown that instability still occurs using the manufacturers' recommended flywheel inertia. It was proposed that tests be conducted to determine the degree of instability and to determine experimentally the necessary flywheel inertia.

6.7 GOVERNOR/GENERATOR COMPATIBILITY

For the electronic load governor, its compatibility with the generator is vital since the governor controls, and is responsible for, the stability of the entire electrical and mechanical system. Depending on the construction of the generator and the voltage regulation system, the two devices may react disadvantageously with each other. The proposed governor transient test will also provide a system for assessing the compatibility of the selected generator with the electronic load governor.

6.8 ELECTRONIC LOAD GOVERNOR TRANSIENT STUDY

The electronic load governor compatibility and stability was tested on an experimental rig shown in Figs 6.9 and 6.10, with the following objectives:-

1 to determine the compatibility of the 6 kVA Markon type B21D brushless synchronous generator with the electronic load governor.

2 to study the transient behaviour of the electronic load governor.

3 to determine the correct flywheel inertia to be used.

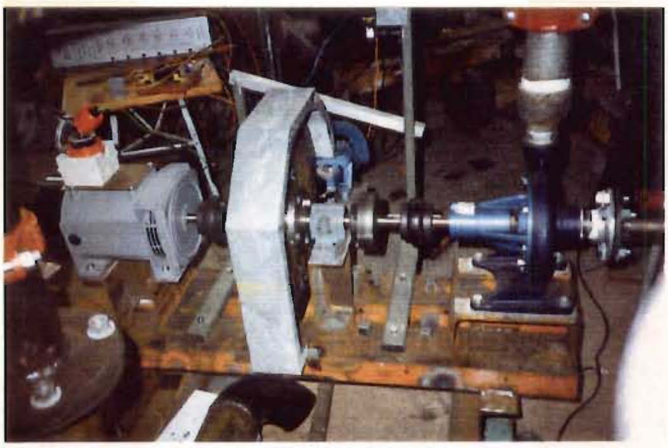


Fig 6.9 The governor transient test rig

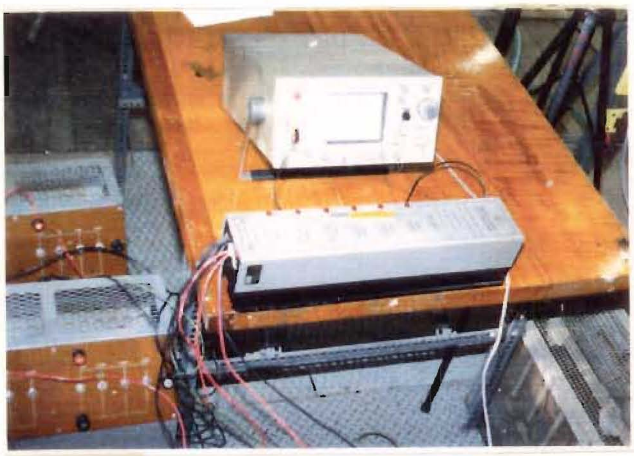


Fig 6.10 The electronic load governor and monitoring equipment

The test rig comprises a centrifugal pump model Davis 2M, directly coupled to the Markon B21D brushless generator. An oscilloscope model Trio 60 MHz CS-1060, was used to monitor the wave-form and the frequency of the electrical system.

Five sets of tests were conducted with different sizes of flywheel to determine the optimum configuration:-

Test number	Inertia (kgm^2)	Multiple of recommended inertia, test 2	Diameter (mm)	Mass (kg)
1	0	no flywheel	0	0
2	0.11	1 time	330	7.3
3	0.23	2 times	406	11.0
4	0.41	4 times	470	14.7
5	0.64	6 times	470+406	24.0

Table 6.1 The flywheel design data

NOTE: Flywheel inertia of 0.11 kgm^2 (Test No 2) represents the inertia recommended by the governor manufacturer.

The system ballast transient response occurs due to the switching between the load steps within the ballast loads. This transient response is small compared with the transient response caused by the consumer suddenly applying a load. It can therefore be assumed that the small transient within the ballast is satisfied once the large transient caused by the consumer is satisfied.

a) Results presentation

The test results are shown in Table 6.2. The set-up was first tested without the draft tube or flow straightener. Instability within the turbine was obvious. It was possible to notice surging effects and swirling of the fluid leaving the turbine. After the draft tube and flow straightener was fitted, the system operation appeared more uniform with increased power output as recorded by the ammeter.

The transient response test was done by instantaneously switching on a 2 kw load at the consumption point, thus causing a sudden drop in the generator speed in response to the load changes.

The resulting overshoot and undershoot in frequency was obtained by observing the waveform on the oscilloscope.

Inertia (kgm^2)	Run No	Inlet pressure kPa	Maximu m (mm)	Overshoot (mm)	Undershoot (mm)
1 (0)	1	690	50	20	10
	2	496	45	35	6
	3	379	45	35	5
2 (0.11)	1	690	45	15	5
	2	496	45	15	5
	3	379	35	15	10
3 (0.23)	1	690	50	6	4
	2	496	50	6	4
	3	379	50	6	3
4 (0.41)	1	690	45	5	0
	2	496	45	5	0
	3	379	45	5	0
5 (0.64)	1	690	45	0	0
	2	496	45	0	0
	3	379	45	0	0

Table 6.2 The transient test results

The graphical representation of the results are shown in Fig 6.11. The graphs were plotted using an arbitrary time scale since it was not possible to determine accurately the period of oscillation.

The figures show the characteristic of a spring damper mechanism. A critically damped case is desired.

The system with no flywheel is very unstable due to lack of inertia on the shaft. During the test, it was possible to see and hear the fluctuation in speed as the load switching occurs within the ballast resistors. This confirmed the need for a correctly sized flywheel for both of the transient responses. This is represented by curve 1.

Curve 2 illustrates the transient response with flywheel of inertia 0.11 kgm^2 as recommended by the governor's manufacturer. The results showed that the transient has overshot past the bandwidth of the governor, which implies that the system is still underdamped. Increased in shaft inertia would be required.

A flywheel of inertia twice the recommended inertia is added and 1.5 kW load is switched on at the consumption point producing a transient response represented by curve 3. The results showed that the damping has improved but the system is still not critically damped.

Curves 4 and 5 illustrate the transient response with flywheel inertia of 4 and 6 times the recommended inertia respectively. The results showed that for curve 4, the system is approximately critically damped. Further increase in inertia would results in an even more stable system, but there is a compromise between the system stability and the mechanical design.

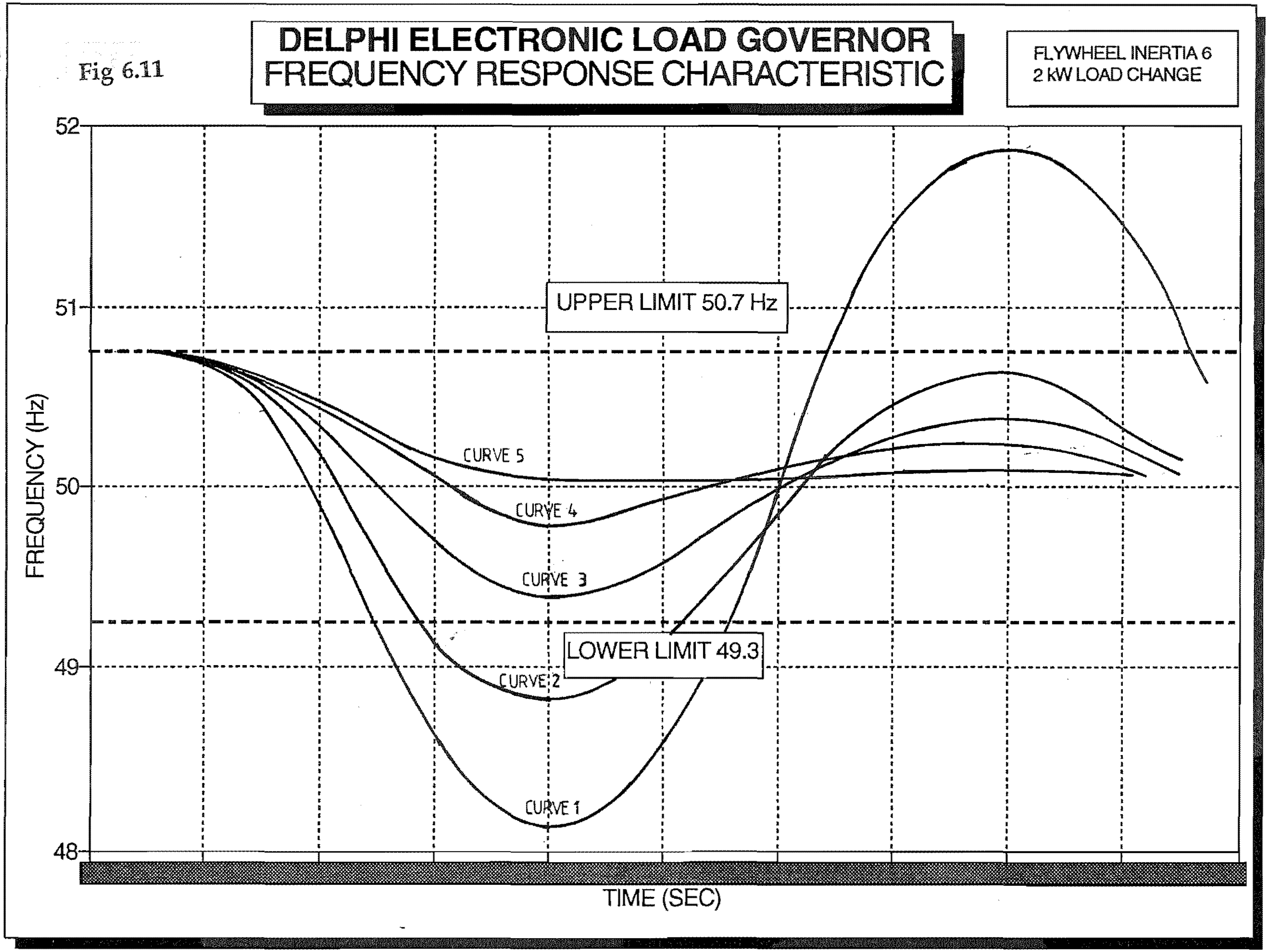
6.9 ALTERNATIVE METHOD OF TRANSIENT TEST

Although the method of studying the transient response of the governor was adequate for this application, an alternative method was explored. This method attempted to record the change in frequency as an analogue voltage by use of a frequency to voltage converter. The zero-crossings of the generated waveform were converted to pulses, these pulses were then counted and an analogue voltage produced proportional to the number of pulses. The data was transferred into a computer through an analogue to digital converter for analysis. The results produced were difficult to analyse due to the poor quality of the data acquisition system.

6.10 CONCLUSION

Based on the test results, it can be concluded that a flywheel of inertia of at least four times the recommended inertia is required for the Delphi electronic load governor. Furthermore, the electronic load governor was proven to be compatible with the Markon generator AVR system.

Fig 6.11


DELPHI ELECTRONIC LOAD GOVERNOR
FREQUENCY RESPONSE CHARACTERISTIC
FLYWHEEL INERTIA 6
2 kW LOAD CHANGE
52
51
50
49
48
FREQUENCY (Hz)
UPPER LIMIT 50.7 Hz
LOWER LIMIT 49.3
CURVE 5
CURVE 4
CURVE 3
CURVE 2
CURVE 1
TIME (SEC)

CHAPTER SEVEN

7 ELECTRICAL SAFETY AND PROTECTION SYSTEM

7.1 ELECTRICAL PROTECTION SYSTEM

The microhydro scheme is to operate on a run-of-the-river principle, generating a constant base load power of 4.8 kW at 0.8 power factor. During maximum demand period, this can be easily overloaded. Similarly, the mechanical power to the generator may fall below the design value due to a reduction in water supply with the same consequent overloading effect. Moreover, microhydro plants are usually located in mountainous areas of high rainfall, and high humidity. In such localities they are vulnerable to malfunctions. It is therefore essential that adequate protection of the generating equipment and consumer appliances is provided.

7.2 PROTECTION PRINCIPLES

Microhydro power, unlike the urban power supply, has a relatively lower power capacity than the load system to which it is connected. The supply will be from an automatic, unattended source in which the voltage, frequency and current can vary independently and in various combinations outside the normal supply limit. These variations can be the result of a malfunction in any part of the complete scheme.

Basically the protection principles are to sense voltage, frequency, and current separately and provide an appropriate automatic protective response. A fail-to-safe condition is required. General layout of the electrical protection system is shown in Fig 7.1.

As outlined by Bryce and Giddens [3], the approach to the design of system malfunction protection equipment depends on:-

1. The designer's responsibility to the consumer
2. The requirement of any statutes and regulations
3. The skill that will be available for recognising and responding to a malfunction
4. The system design life
5. The facilities available for maintenance and repairs
6. Cost

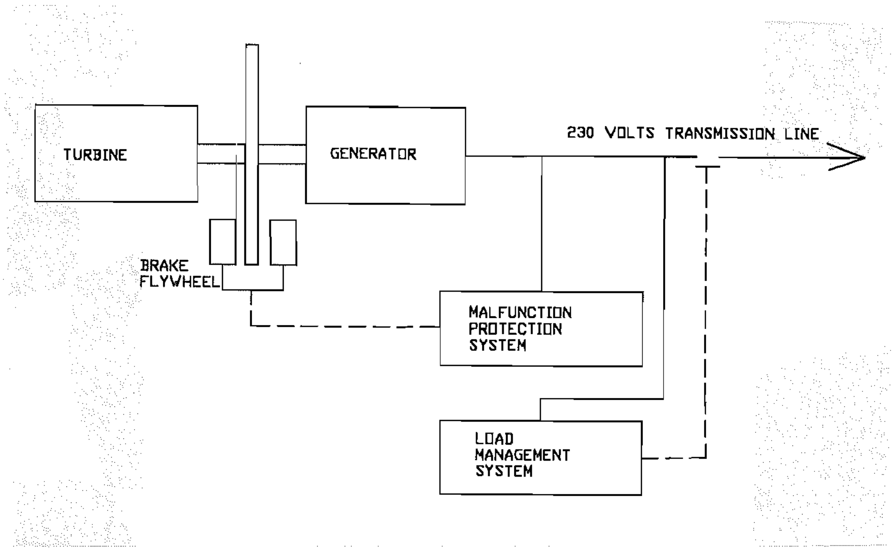


Fig 7.1 Electrical protection system

There are four levels of protection which cover the fault states that may occur. These are characterised by the degree of isolation from the consumer.

1 Consumer circuit protection
2 Consumer mains protection
3 Transmission line
4 Generator

7.3 FAIL-SAFE MALFUNCTION PROTECTION SYSTEM

The philosophy is to design a braking or an isolating system that brings the machine to a safe condition in the event of a malfunction without the aid of external energy, either electrical or mechanical. This may be achieved in two ways:-

i) Hydraulic Energy Diversion

This can be done by diverting the water from the turbine with an auxiliary by-pass, which eventually allows the turbine to stop.

ii) Mechanical braking

This is achieved through mechanically stopping the machine shaft by means of a mechanical braking system.

Both of these methods have been proved possible. However, the former is not recommended as any attempt to control the high head inlet water can cause serious water hammer effects which may lead to pipe rupture. This method also suffers from a slow response. The latter on the other hand possesses the following advantages:-

1 Rapid response.

2 The availability of the disc flywheel to provide inertia means that it can also serve as a disc for the brake callipers.

3 The brake can also be used during normal stopping and starting procedures.

4 The hydraulic impedance of the centrifugal pump operated as a turbine does not cause any serious water hammer effect when it is rapidly stalled.

Mechanical layout of the braking system is covered in Chapter 8. Basically, a power solenoid is used to latch a set of brake callipers held open by tension springs. When malfunction occurs, power to the solenoid is disabled causing the brake to actuate under the tension of the springs. The machine will then be brought to a stop.

a) Systems operation

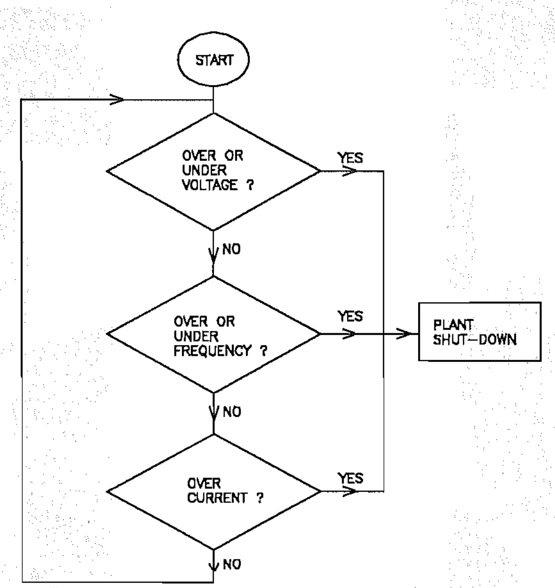


Fig 7.2 Flow chart of possible malfunctions

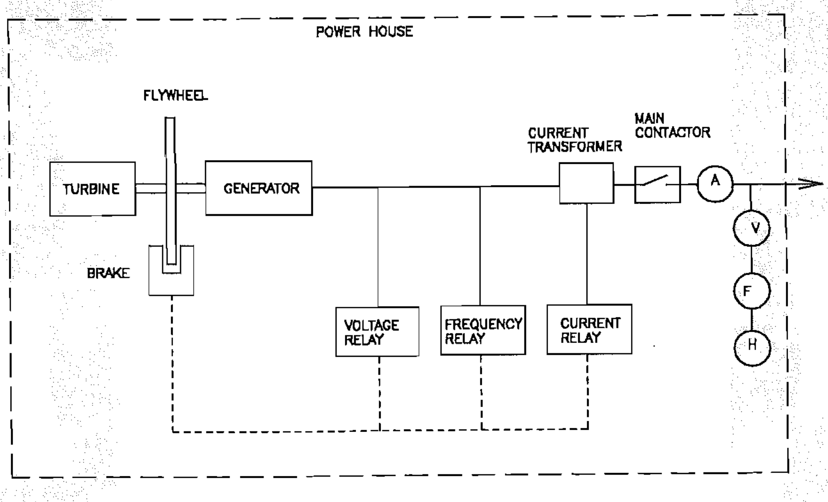


Fig 7.3 Malfunction protection system

i) Over/Under Voltage

It is necessary to protect the connected equipment against extreme voltages. e.g. lights, television set, may be damaged by high voltages whilst low voltages can cause high current in motors. For the generator, extreme voltage can be caused only by serious overspeeding or failure of the voltage regulator. In either case, plant shut-down is appropriate and this can be initiated with an under/over voltage relay. The voltage relay requires the voltage to be maintained within the range of 180 v to 260 v.

ii) Over/Under Frequency (speed)

The function of the load governor is to hold the frequency and voltage constant within acceptable limits. However, overspeed can still occur due any of the following malfunctions :-

1 Load rejection
2 Loss of excitation
3 Failure of automatic voltage regulation
4 Failure of ballast loads
5 Failure of the load governor

In this case a frequency relay is used and is set to lower and upper limits of 45 Hz and 55 Hz. If a fault is detected, system shut down is necessary. For low to medium specific speed machines (centrifugal pumps) such procedures impose no dangerous water hammer on the penstock. This is based on the test conducted on the microhydro prototype outlined in Chapter 6.

If the plant had been stopped as a result of a load rejection, and the fault had not been diagnosed, an over-speed could occur on starting while the brake was being manually held open. This feature should be safeguarded by the over speed relay disabling the manual override.

Overspeed can also occur due to loss of excitation. This means that no voltage is generated. In this case the over/under frequency relay will not work. However, should a loss of excitation occur, the voltage will drop below the lower limit of the under/over voltage relay and that will result in the shutting down of the plant. Since the failsafe brake is powered from the generator, loss of excitation results in loss of power. This automatically releases the brake.

iii) Over current

Over current protection will be necessary to prevent excessive energy dissipation that would endanger plant or personnel as a result of equipment malfunction. The use of rewirable fuses is likely to be ineffective because the system supply will very likely not have the excess capacity to open them. In such a case, a sensitive over current relay is used. Again, shut down is necessary in the event of such a fault. A user adjustable current setting is required. At full output of 4.8 kW at 0.8 pf, the current is about 26 amps.

Over/Under voltage, over/under frequency and over current protection equipment will be located in the power house. In the event of a malfunction, the protection relays are required to trigger a fail-safe brake and stall the machine. This can be achieved by using the relay to hold open a solenoid which, when de-energised, allows the brake to act. Layout of the system is shown in Fig 7.3.

7.4 LOAD MANAGEMENT SYSTEM

As outlined in section 2.5, overloading can easily, and not infrequently occur. This is only to be expected if, in fact, the consumer is trying to make full use of the resource. Similarly, the mechanical power to the generator may fall below the design value due to a reduction in the water supply to the turbine with the same consequence of overloading of the generator. This situation does not call for immediate isolation but the consumer needs to make a load adjustment. An alarm, warning of the need to shed load, followed by isolation after a time interval, is required.

Recall from Chapter 6 that the frequency droop of the load governor is over the range of 49.3 Hz to 50.7 Hz. This implies that when the consumer is using the full output from the generator, the frequency will be at 49.3 Hz. On the other hand, if there is no consumer load, and the governor is switched to full dummy loads, the frequency will be at 50.7 Hz. A frequency relay is used to determine when overload occurs, i.e when the frequency drops below 49.3 Hz. A frequency drop of 2 Hz from the nominal value of 50 Hz is used to detect this condition. An alarm system and a time delay are also used. Layout and operational features of the load management system are shown in Fig 7.4 and Fig 7.5.

Like the electronic load governor, the overload relays and timer system may be located anywhere on the reticulation since frequency is constant throughout the entire system. However, it would normally be located at the consumer end of the reticulation to enable load use to be monitored.

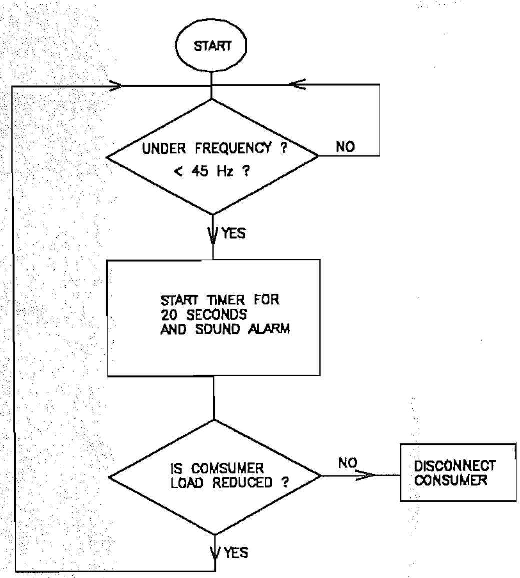


Fig 7.4 Flow chart of load management system

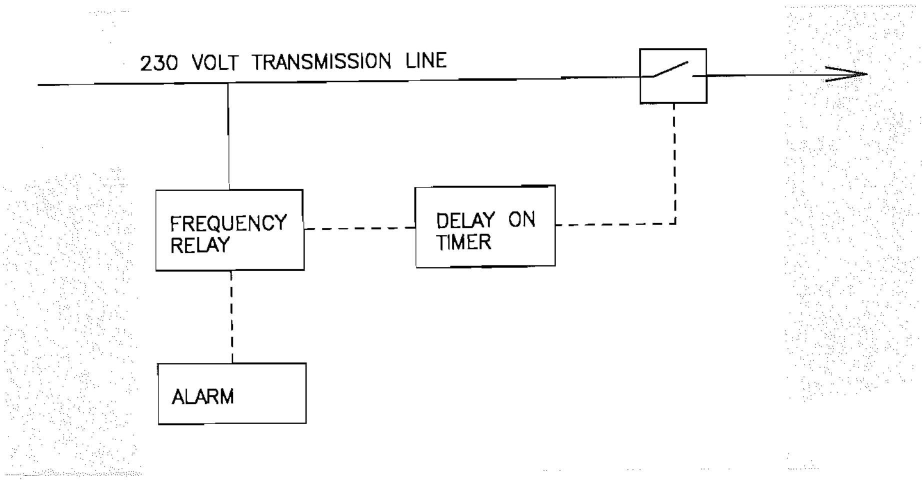
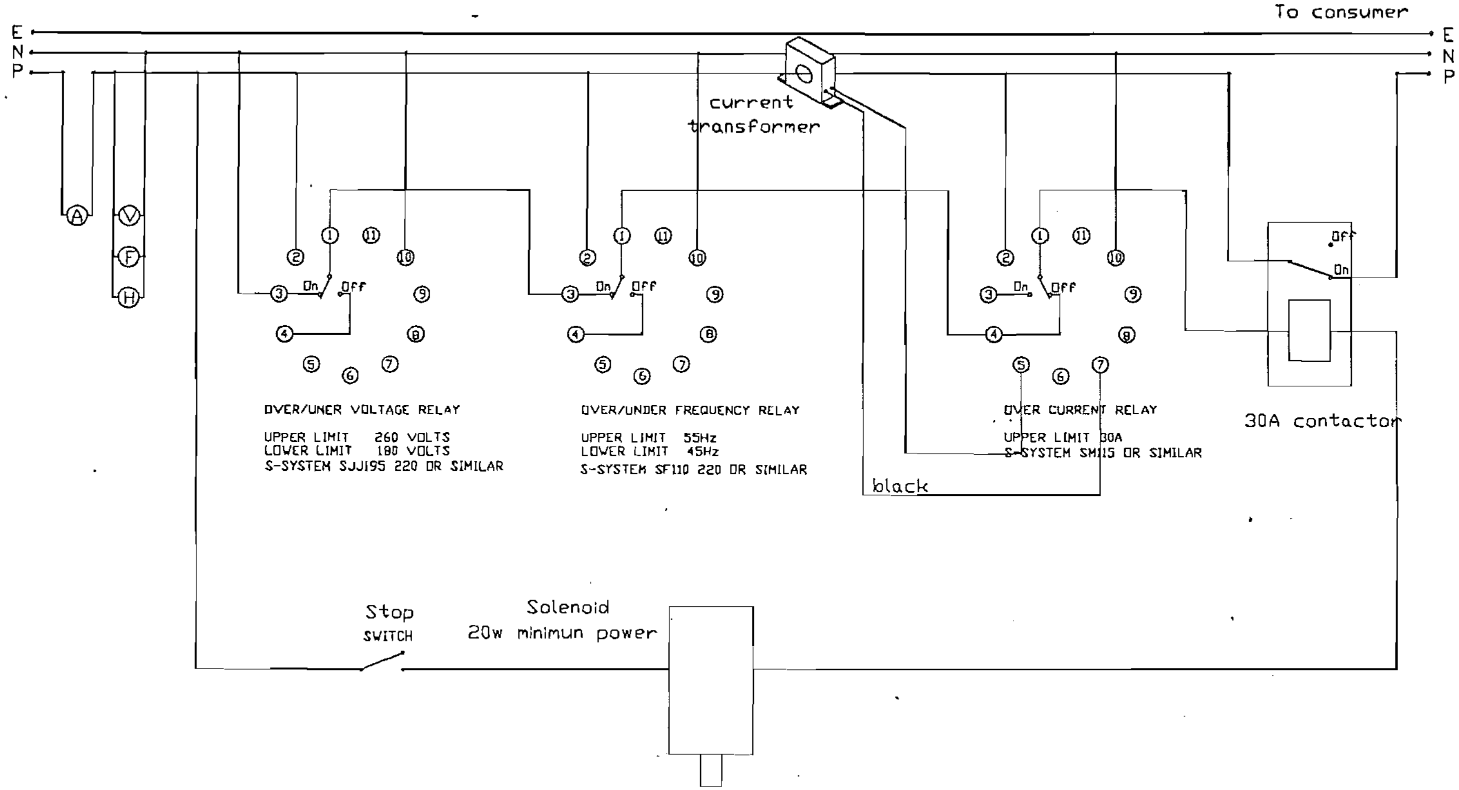


Fig 7.5 Load management system

7.5 DETAIL DESIGN

Detail design of the Malfunction protection system and the Load management system are outlined in Figs 7.6, 7.7, 7.8, 7.9.

Fig 7.6



ITEM	DESCRIPTION	QNTY

MATL:

THIRD ANGLE PROJECTION A3

TOLERANCES:

unless otherwise stated

MALFUNCTION PROTECTION SYSTEM CIRCUIT

SCALE: NTS

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S . HENG
CHKD:
DATE: 5 JUNE 90

DRG No: D610

Fig 7.7

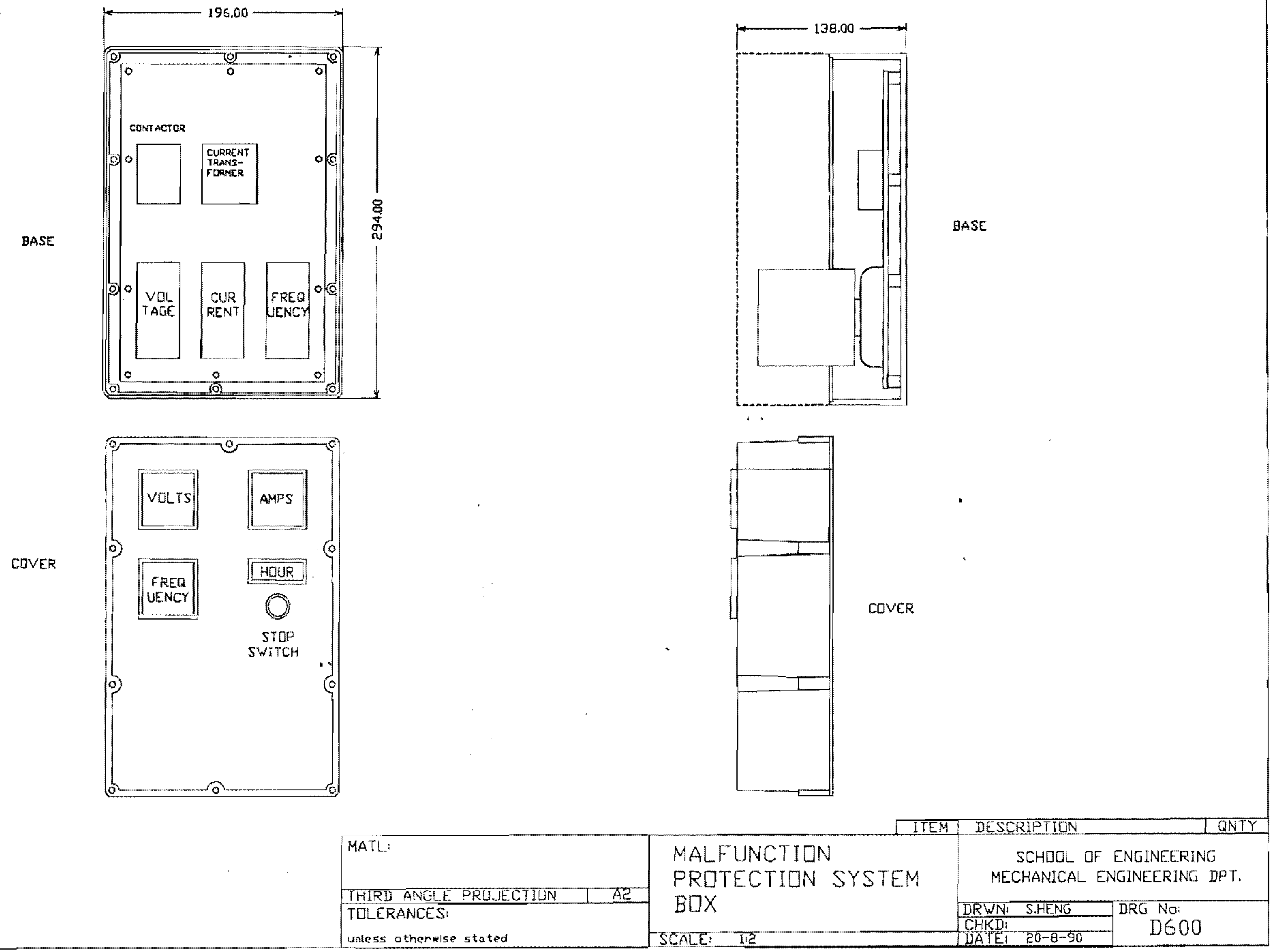
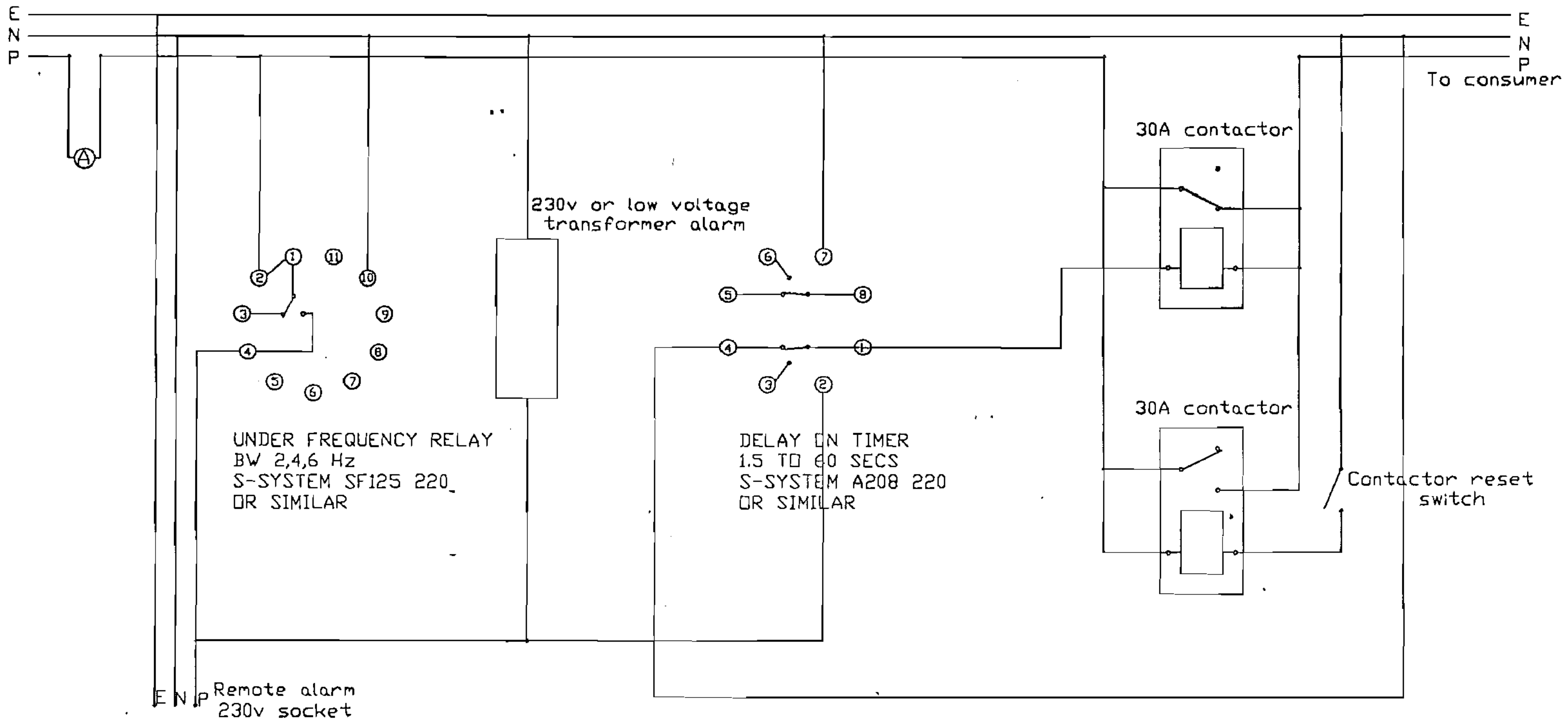

196.00
294.00
138.00
BASE
CONTACTOR
CURRENT TRANS-FORMER
VOL TAGE
CUR RENT
FREQ UENCY
BASE
COVER
VOLTS
AMPS
FREQ UENCY
HOUR
STOP SWITCH
COVER
ITEM
DESCRIPTION
QNTY
MATL:
THIRD ANGLE PROJECTION
A2
TOLERANCES:
unless otherwise stated
MALFUNCTION PROTECTION SYSTEM BOX
SCALE: 1:2
SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.
DRWN: S.HENG
CHKD:
DATE: 20-8-90
DRG No:
D600

Fig 7.8



ITEM	DESCRIPTION	QNTY

MATL:

THIRD ANGLE PROJECTION A3

TOLERANCES:

unless otherwise stated

LOAD MANAGEMENT
SYSTEM CIRCUIT

SCALE: NTS

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

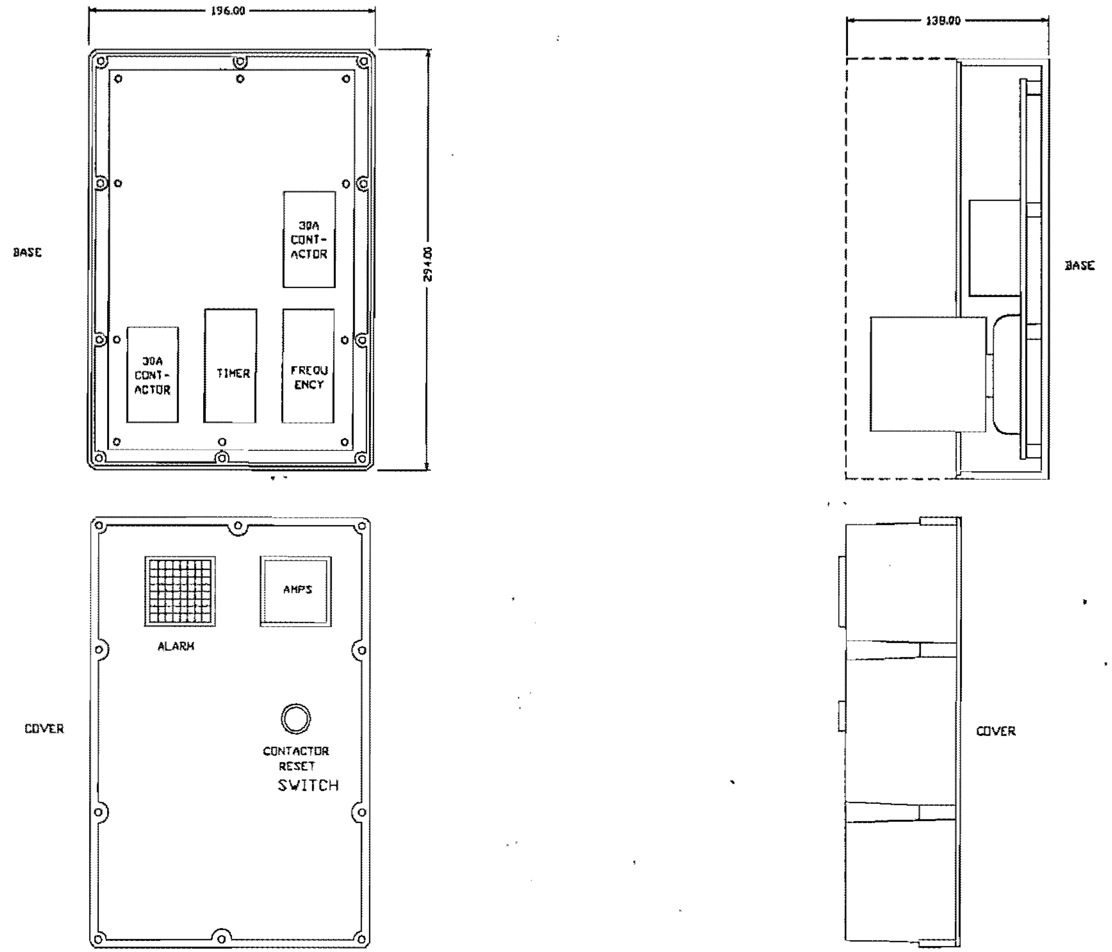
DRWN: S . HENG

CHKD:

DATE: 5 JUNE 90

DRG No:
D630

Fig 7.9



ITEM	DESCRIPTION	QNTY

MATL:

THIRD ANGLE PROJECTION A2

TOLERANCES:

unless otherwise stated

LOAD MANAGEMENT SYSTEM BOX

SCALE: 1:2

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 3-9-90

DRG No: D620

CHAPTER EIGHT

8 THE 5 kW MICROHYDRO GENERATING SET

8.1 OVERVIEW

The need for standard urban quality electricity in isolated localities has led to the design and development of the 5 kW microhydro generating set. The concept behind the scheme was maximum standardisation by the use of locally available proprietary components. The advantages of the applied concept are readily available parts and interchangeability which have the overall effect of reducing the complexity and cost of manufacture.

Extensive research and development have been conducted on various parts making up the complete microhydro scheme. The introductory Chapters (1 to 3) look at the feasibility of the schemes, while Chapters 4 to 7 look at the detail design aspects of the turbine, generator, load governing and the safety and malfunction protection system. In this Chapter, detailed design of the complete microhydro generating system is described. Use was made of a Computer Aided Design (CAD) package called Intergraph Microstation, and a Finite Element Stress Analysis (FEA) package called Algor. Stress analysis using standard procedure is also covered.

8.2 GENERAL MECHANICAL LAY OUT

a) The 5 kw microhydro generating set general description

The complete 5 kW microhydro generating set assembly is shown in Fig 8.1 (See page 88). The turbine is directly coupled to the synchronous generator via a flexible rubber coupling and spacer. In essence, the set up is a standard pump motor set with the synchronous generator in place of the induction motor. Mounted on the spacer is a disk flywheel of diameter 470 mm which provides extra inertia of 0.41 kgm^2 to the rotating shaft. Layout of the generator/turbine coupling is shown in Fig 8.2 (See page 89). The flywheel is mounted on the shaft supported by the turbine bearings.

b) Pipework and valves

Water to the turbine is supplied through pipework and valves as shown in Fig 8.3 (See page 90). The pipework and flanges are designed in accordance with BS 4772. There are two sizes of pipework for the 5 machines, 100 mm and 150 mm. The larger pipe size was necessary to minimise losses in the penstock for the machines with the largest flow.

c) Draft tube

As outlined in Chapter 4, a draft tube is a necessary requirement for the microhydro generating set. Not only does it increase the efficiency of the machine, but a draft tube with a flow straightener will also stabilise the discharging water by eliminating the swirl component of the flow. Draft tube for a discharge flange diameters of 50 mm, 65 mm and 80 mm are shown in Fig 8.4 (See page 91). Since the draft tube is fitted on the low head side, it can be constructed out of 2 mm rolled mild steel sheet.

The configuration of the draft tube based on a design guide for a diffuser [50] is designed to provide maximum velocity head recovery at minimum friction loss and head losses, due to separation effect. (See calculation in section 8.4).

d) Baseplate

The baseplate is constructed out of a single piece of RHS to minimise structural distortion during installation. Detail of the baseplate is shown in Fig 8.5 (See page 92).

8.3 DETAILED MECHANICAL DESIGN

a) Shaft stress calculations

The turbine rotor assembly comprises the shaft, impeller, sleeves, bearings, and other components such as mechanical seals that rotate as a unit. The primary component of the rotor assembly is the shaft. The turbine shaft transmits energy from the impeller to the generator. This section will be concerned primarily with shaft stresses for steady state and transient conditions. In practice, it is not necessary to calculate stresses in the pump shaft if the application is for pumping. For turbining however, shaft stress calculations are necessary due to the unusual operating conditions as outlined below.

1. Reversed direction of shaft rotation
2. Reversed direction of water flow
3. Increases in head and capacity to approximately twice the pumping case.
4. Introduction of additional stresses on the shaft due to deadweight of flywheel, and impact stresses due to sudden braking.

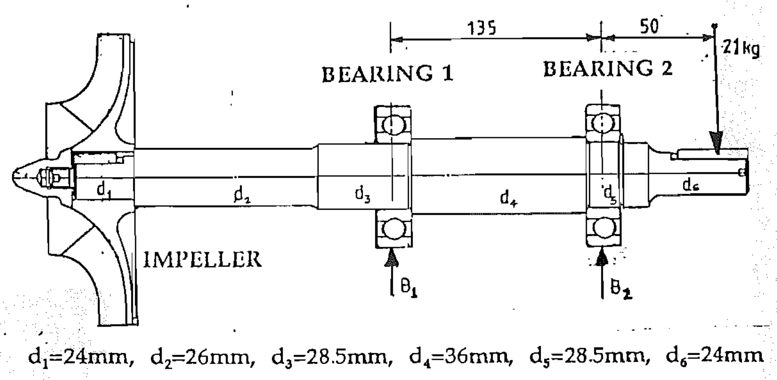


Fig 8.6 The turbine shaft

The shaft material is Stainless steel 431 S29, with the following mechanical properties.

$F_u = 900$ MPa
$F_y = 680$ MPa
$F_c = 635$ MPa
$F_e = 259$ MPa

The shaft will be subjected to,

1 Tension
2 Compression
3 Bending
4 Torsion

For life time loading conditions of more than 1000 cycles, fatigue analysis is necessary. Therefore, the forces due to the weight of the flywheel have an effect on the fatigue life of the shaft. From Labanoff [42], the equation for acceptable fatigue loading without axial load is,

$$\left(\frac{\tau_m}{.5nF_y}\right)^2 + \left(\frac{\sigma_a}{nF_e}\right)^2 \leq 1 \qquad \text{(E8.1)}$$

From Section A4.2, Appendix 4, the following results was obtained,

$\therefore \quad \sigma_a = 3.88$ Mpa
$\tau_m = 3.9$ MPa

Inserting these values in equation E8.1, it is found that $0.0008 \leq 1$ as required.

The primary loads of the turbine shaft are generally torsion and bending loads. The impeller is arranged to minimise axial thrust, and no axial load will be assumed for this application. A safety factor of 1.5 is applied to F_e and F_y to account for unanticipated loads. Equation (E8.1) is applied at the location(s) where stresses are the highest.

The flywheel can safely be overhung without risk of fatigue failure. More detailed analysis is shown in Appendix 4, Section A4.1.2.

Stresses in the shaft have been further analysed in a computer finite element analysis, the results being shown in Fig 8.7.

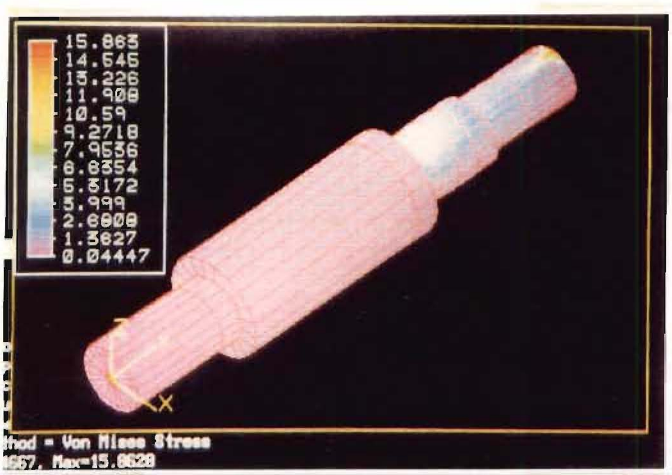


Fig 8.7 Stresses in the turbine shaft due to weight of the flywheel

b) Forces and torque during braking

The life time loading conditions of less than 1000 cycles can be considered as static loads with no fatigue life. Forces and moments during braking are considered to occur less than 1000 times for the life time of the machine. Therefore it is necessary to analyse the pump shaft for static strength only. Again, there is no axial stress for this application. More detailed analysis is outlined in Section A4.1.3, Appendix 4.

Torsional stress,

$$\tau = 26 \text{ MPa}$$

Bending stress,

$$\sigma = 12.6 \text{ MPa}$$

Von Mises stress

$$\sigma' = \sqrt{(12.6^2 + (3)(26)^2)}$$
$$= 47 \text{ MPa}$$

The stresses in the shaft are well below the maximum allowable stress limit.

Fig 8.8 shows stress analysis done using Algor finite element analysis. Note that the Von Mises stress is compatible with the above calculations.

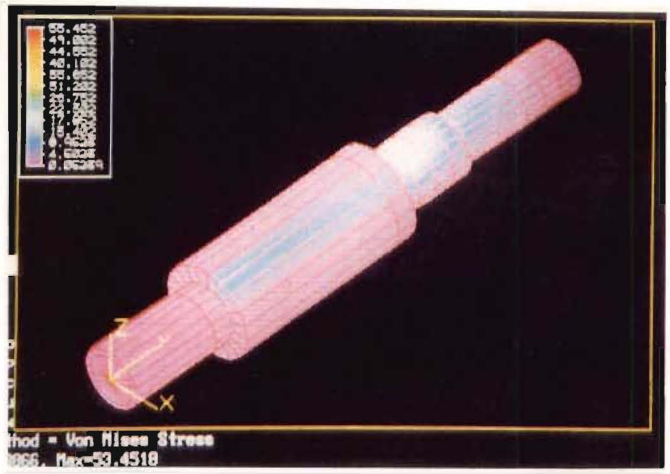


Fig 8.8 Stress in the shaft during the braking operation

c) Stresses in the flywheel

It is necessary to calculate the stresses in the flywheel to check if stresses are within the acceptable limit.

Maximum tangential stress occurs at $r = r_i$ (hub)

For the 5 kW microhydro generating set,

Disk diameter	=	470 mm
r_o	=	235 mm
I	=	0.41 kg/m^2
Inner diameter	=	80 mm
r_i	=	40 mm

Therefore, maximum radial stress,

σ_{rmax} = 12 MPa

Maximum tangential stress occurs at $r = r_i$ (hub)

σ_{tmax} = 47 MPa

Maximum Von Mises stress,

$$\sigma' = \sqrt{(12^2 + 47^2 + (12)(47))}$$

$$= 54 \text{ MPa}$$

This is well within the limit of yield stress for mild steel of 250 MPa. Detailed analysis is outlined in Section A4.2 page A4.9, Appendix 4.

Algor finite element analysis of the flywheel rotating at 3000 RPM is shown in Fig 8.9. It can be seen that maximum stress occurs at the centre of the flywheel. Also the actual flywheel has a centre hole. It was not possible to model this on the computer. The value of stresses was calculated above.

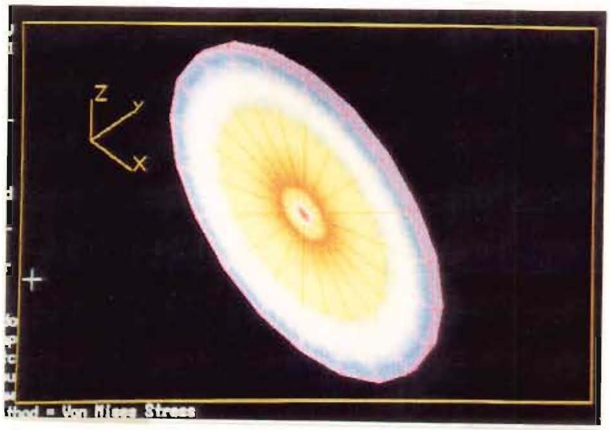


Fig 8.9 Von Mises stress in the flywheel at 3000 RPM

d) Bearing life estimate

Turbine bearings are,

SKF 6303R (Outer)
SKF 6303 (Inner)

Load rating	C(dynamic)	21.6 kN
	C(static)	14.6 kN
	Limiting speed	9000 RPM

Design for one year continuous running (8736 hours)

$$L_n = \frac{10^6}{60N}\left(\frac{C}{p}\right)^p \qquad \text{(EA4.5)}$$

Where:-

N	3000 RPM
C	21.6 kN
p	3 for ball bearings
L_n	Bearing life

$$8736 = \frac{10^6\,(21600/p)}{(60)(3000)}$$

$$p = 1857\text{ N}$$

This limits the radial force to 1.86 kN. At the bearing, the shaft has a steady load of 282 N, and the highest load during braking is 669 N, after allowance has been made for a shock factor of 3.

Therefore, the bearing will last more than one year's continuous operations. However, it is recommended that the bearing is replaced annually.

e) Brake release mechanism

Based on the manufacturer's data sheet, the following detail was obtained for the disc brake calliper. The design of the brake release mechanism is shown in Fig 8.10 (See page 93).

Torque	47 Nm
Disc diameter	470 mm
Mean diameter of brake pad action	411 mm
Force on lever	120 N

Based on the mechanical layout and the above data, the tension spring required to stall the machine has the following characteristic,

Initial length	180 mm
Spring constant	1.1 N/mm

f) Critical speed

If, for any reason, the centre of gravity of the rotating system does not lie on the axis of rotation, the phenomenon of resonance will occur when the shaft revolves with a periodicity that approximates to one of the natural periods of transverse vibration. This state of resonance commonly known as Critical speed or Whirling speed can causes serious consequential damage to bearings due to increased shaft deflection and increase vibration. For this reason, the critical or whirling speed should be avoided as far as possible in normal operation of the machines.

Critical speed is determined by the deflection of the shaft under its own mass or added mass (in this case, the flywheel).

The less the deflection, the higher the critical speed. It is desirable to have the critical speed as high as possible.

$$N_{crit} \propto 1/\sqrt{\delta}$$

For long slender shaft	N_{crit}	Low
For large diameter shaft	N_{crit}	High

From the Algor finite element analysis result (See Fig 8.7), with flywheel weight of 21 kg (i.e. weight of flywheel and coupling hanging on the stub shaft), the shaft deflection under this weight is 0.0091 mm.

By definition, critical speed is,

$$N_{crit} = \frac{946}{\sqrt{.0091}}$$

$$= 10{,}000 \text{ RPM}$$

Therefore, with the operational speed of the machine at 3000 RPM, it is safe from the critical speed of 10,000 RPM.

8.4 HYDRAULIC DESIGN

a) Pipework pressure losses

Head loss results within the pipework are summarised as follows. Calculations are shown in Section A4.4, Appendix 4.

Machines	Velocity (m/s)	Head loss (m)	δh/h (%)	Pipes size (mm)
50x32-200	1.1	0.6	0.4	100
65x40-200	1.7	1.4	1.7	100
65x50-160	1.7	1.4	2.1	100
80x65-160	1.2	0.6	1.3	150
80x65-125	1.5	1.0	3.4	150

Table 8.1 Head loss in the pipe

b) Draft tube design

See calculations in Section A4.6, Appendix 4.

Machines	Q (l/s)	Inlet velocity (m/s)	Head loss (m)	Inlet pipes size (mm)
50x32-200	10	3.0	0.08	100
65x40-200	14.5	4.4	0.18	100
65x50-160	14.5	4.4	0.18	100
80x65-160	21.4	4.3	0.24	150
80x65-125	28.7	5.7	0.41	150

Table 8.2 Draft tube design data

c) Water hammer

It is not possible to make formal decisions on the magnitude of possible water hammer pressures because this project is limited to determining the generating equipment alone, and the configuration of all possible penstocks cannot be assessed and allowed for in advance.

The approach adopted herein is to assume that there will be operating procedures to protect the pipeline from excessive pressures. These procedures must limit the excessive pressure to the maximum test pressure of the penstock pipe material. The maximum test pressure is normally 1.5 times the working pressure of the pipe. The working pressure of the penstock pipe must be at least equal to the full penstock static head.

For the operating case, the penstock sizing would limit the losses. These losses would be unlikely to exceed 35% of the static head. Thus the maximum test pressure of the pipe material will be at least 1.5 times the static head, and the static head can be up to 1.5 times the operating pressure.

The proposed operating procedure for controlling the maximum pressure rise on valve operation is to provide a valve that cannot be closed very rapidly and has a good closing characteristic, i.e. a geared butterfly valve, and to provide a pressure gauge which is marked to show a maximum permissible overpressure limited to twice the normal operating pressure. Clear operating instructions would be required to be placed beside the valve to instruct the operator on how to open and close the valve and not produce excessive pressures.

In fact, in the design of the equipment, the forces will be based on a peak pressure, i.e. design head, due to water hammer of twice the static pressure which will be taken to be 1.5 times the turbine operating pressure.

The penstock pipes are most likely to be plastic and are expected to have working pressures little more than the static pressure of the site.

Machines	Operating head (m)	Design head (m)	Design press (MPa)	Pipe dia(mm)	Force (kN)
50x32-200	155	465	4.56	100	35.8
65x40-200 [185]	80	240	2.35	100	18.5
65x50-160	64	192	1.88	100	14.5
80x65-160 [142]	42	126	1.24	150	21.9
80x65-125	28	84	0.83	150	14.7

Table 8.3 Water hammer pressure forces on pipe mounting design data

d) Preventing cavitation

As discussed in Section 3.8 (E3.24) that, the maximum suction heads to prevent cavitation for centrifugal pumps as turbines is given by,

$$Z_{max} = \frac{p_{atm}}{\rho g} - \frac{p_v}{\rho g} - TREH$$

Where,

TREH = σ_cH

Now,

$p_{atm}/\rho g \quad = 10.3$ m
$p_v/\rho g \quad = 0.24$ m

By using Figs 8.11 and 8.12 to determine the turbining cavitation constant, the following results can be obtained.

Machines	Head (m)	N_{st}	σ_c	TREH (m)	z_{max} (m)
50x32-200	155	0.025	0.012	1.92	8.2
65-40-200[185]	85	0.0475	0.017	1.39	8.7
65x50-160	64	0.075	0.027	1.81	8.3
80x65-160[142]	43	0.12	0.045	2.03	8.1
80x65-125	28	0.202	0.10	2.9	7.2

Table 8.4 Maximum suction head to prevent cavitation at sea level

8.5 ELECTRICITY COST ANALYSIS

For the 5 kW microhydro generating set, the electricity cost estimates are shown below. The analysis assumes the following:

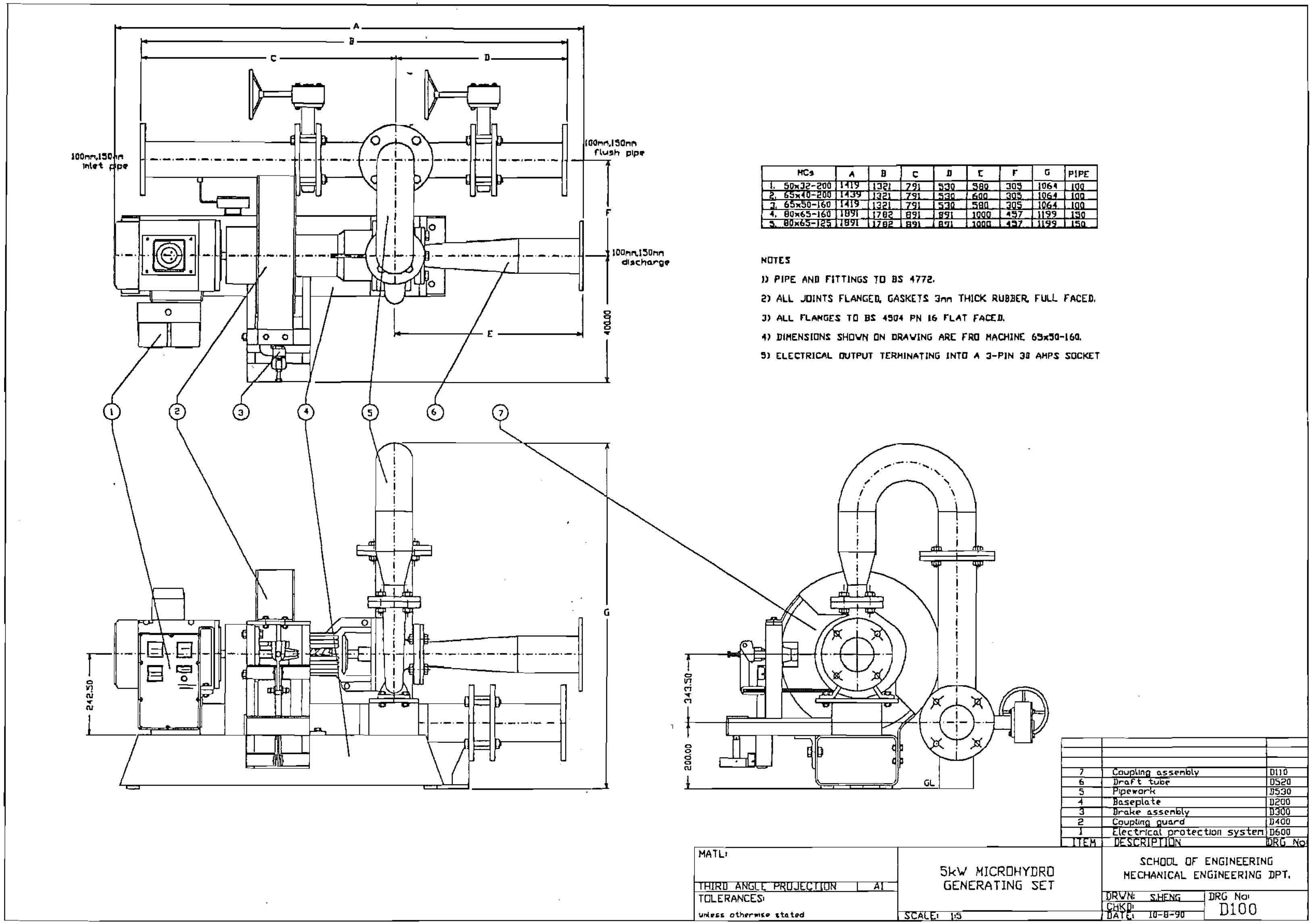
1. Free owner labour for construction
2. A system life of 20 years
3. 10% annual repayment

Machinery and control equipment	$15,000	
Penstock	$10,000	
Incidentals	$5,000	
TOTAL	$30,000	
Present annual value		$3,000
Annual maintenance		$500
Total cost per year		$3,500

Electricity cost at 1.5 kW continuously **27 cents/kWh**
Electricity cost at 4.8 kW continuously **8 cents/kWh**

The energy compares favourably with Standard urban residential supply of between 8 to 12 cents per kWh. Moreover, in rural communities such as the Chatham Islands where all power is from diesel generating sets, the electricity cost is in the order of 40 cents per kWh.

Fig 8.1



MCs	A	B	C	D	E	F	G	PIPE
1. 50x32-200	1419	1321	791	530	580	305	1064	100
2. 65x40-200	1439	1321	791	530	600	305	1064	100
3. 65x50-160	1419	1321	791	530	580	305	1064	100
4. 80x65-160	1891	1782	891	891	1000	457	1199	150
5. 80x65-125	1891	1782	891	891	1000	457	1199	150

ITEM	DESCRIPTION	DRG No:
7	Coupling assembly	D110
6	Draft tube	D520
5	Pipework	D530
4	Baseplate	D200
3	Brake assembly	D300
2	Coupling guard	D400
1	Electrical protection system	D600

Fig 8.2

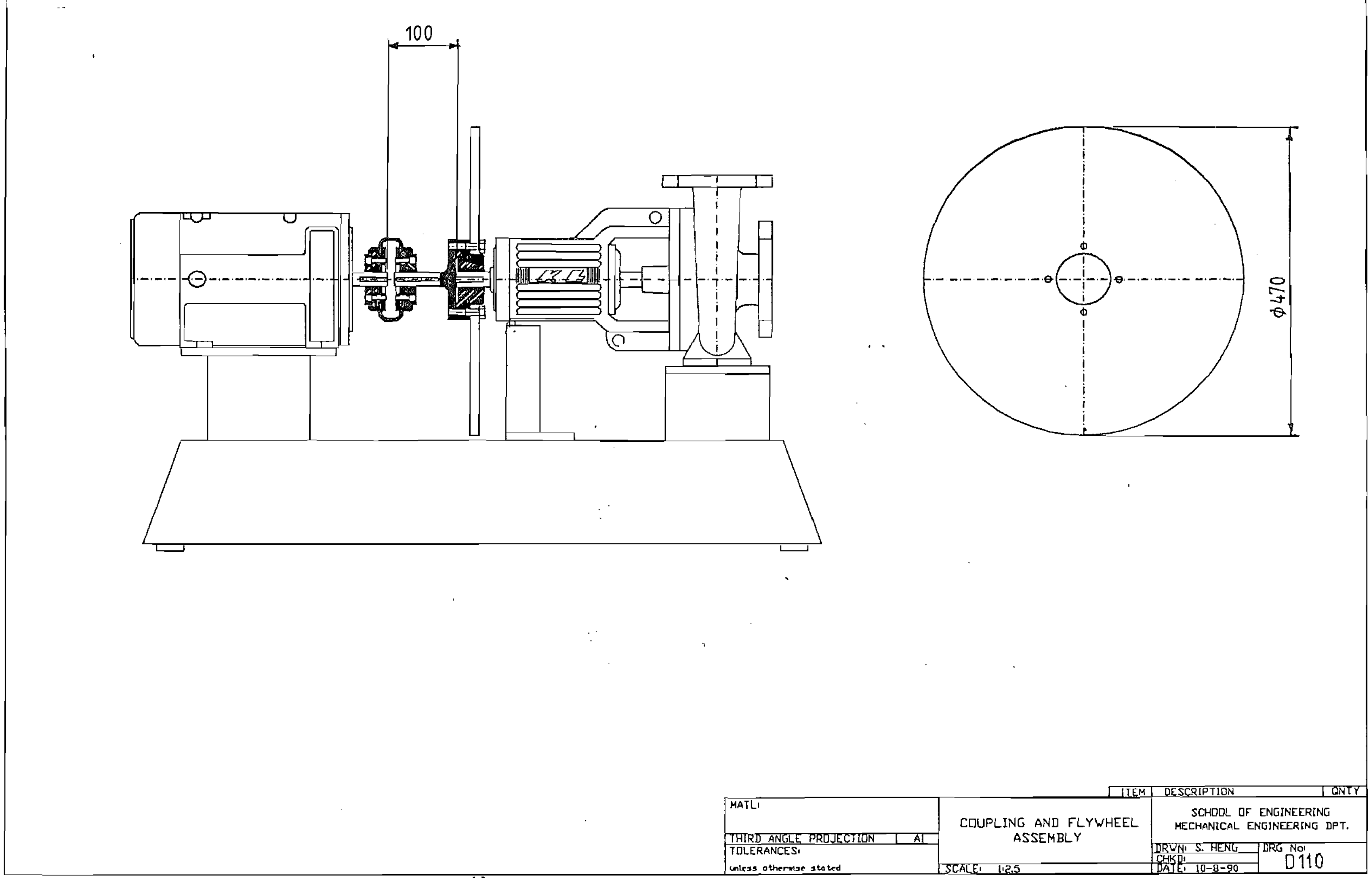
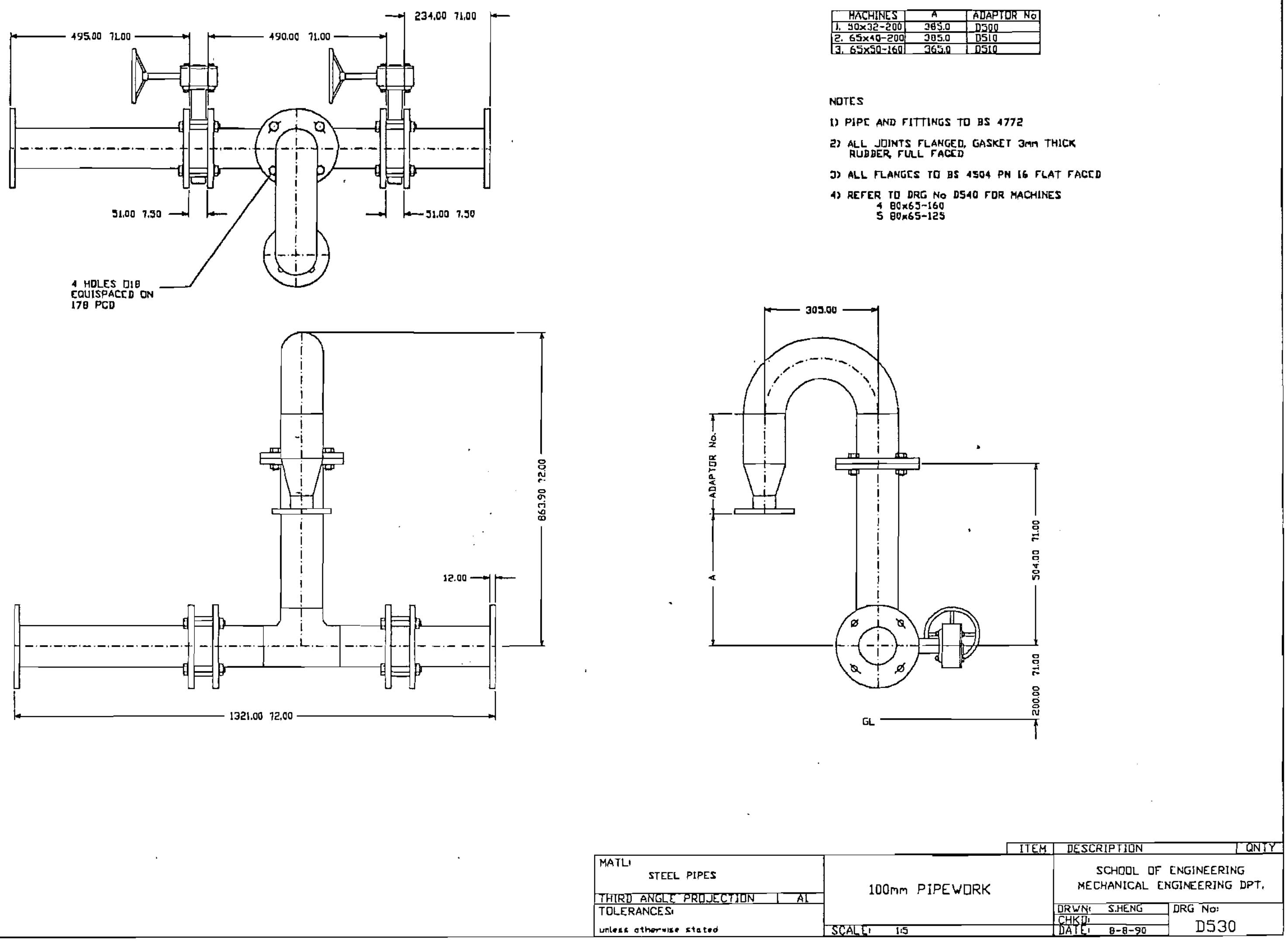
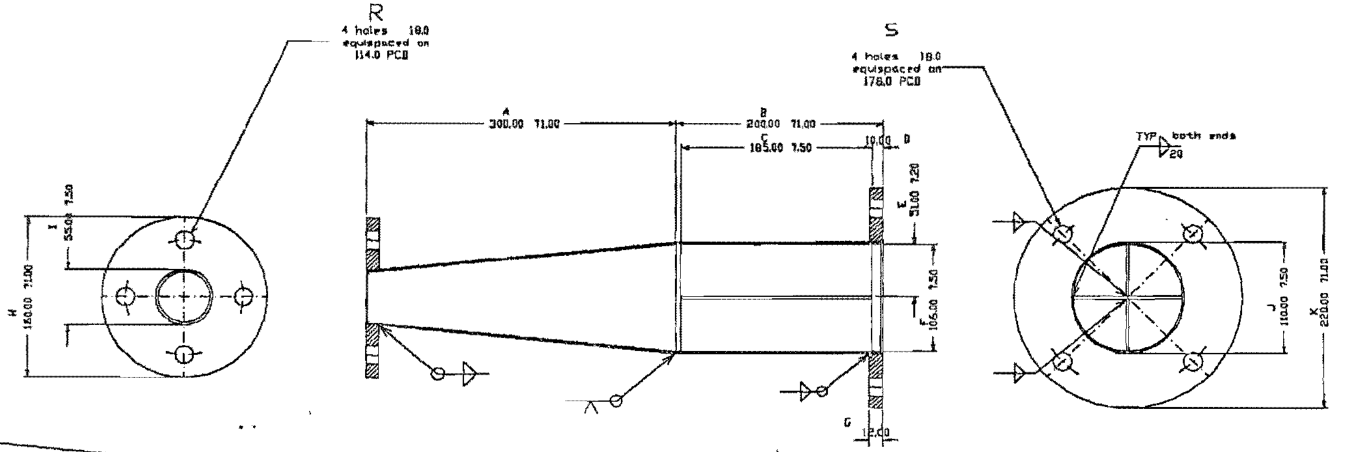

100
φ470
ITEM
DESCRIPTION
QNTY
MATL:
THIRD ANGLE PROJECTION
A1
TOLERANCES:
unless otherwise stated
COUPLING AND FLYWHEEL ASSEMBLY
SCALE: 1:2.5
SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.
DRWN: S. HENG
CHKD:
DATE: 10-8-90
DRG No:
D110

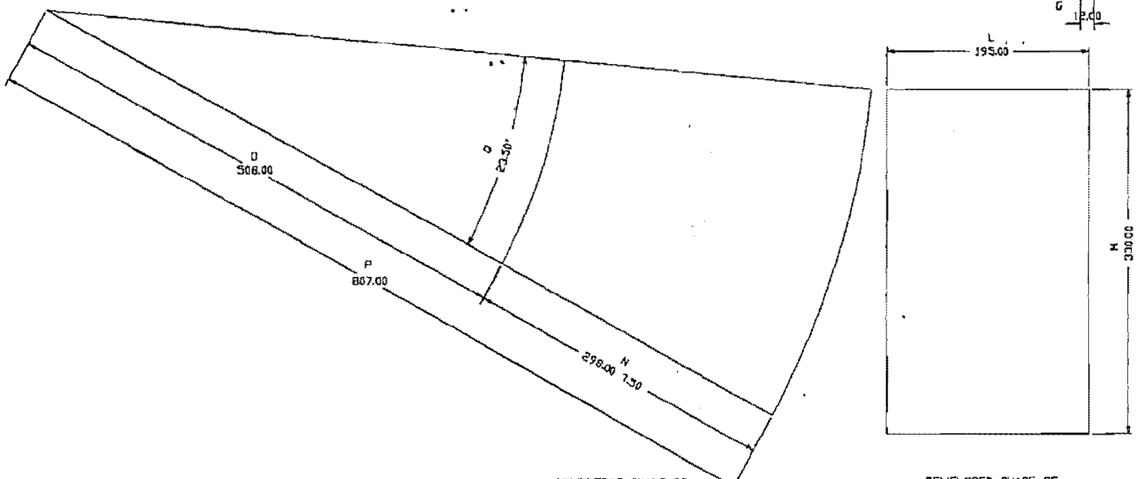
Fig 8.3



495.00 71.00
490.00 71.00
234.00 71.00
51.00 7.50
51.00 7.50
4 HOLES D18 EQUISPACED ON 178 PCD
863.90 12.00
12.00
1321.00 12.00
305.00
ADAPTOR No.
A
504.00 71.00
200.00 71.00
GL
MACHINES | A | ADAPTOR No
1. 50x32-200 | 385.0 | D500
2. 65x40-200 | 385.0 | D510
3. 65x50-160 | 365.0 | D510
NOTES
1) PIPE AND FITTINGS TO BS 4772
2) ALL JOINTS FLANGED, GASKET 3mm THICK RUBBER, FULL FACED
3) ALL FLANGES TO BS 4504 PN 16 FLAT FACED
4) REFER TO DRG No D540 FOR MACHINES
4 80x65-160
5 80x65-125
ITEM
DESCRIPTION
QNTY
MATL: STEEL PIPES
THIRD ANGLE PROJECTION
A1
TOLERANCES: unless otherwise stated
100mm PIPEWORK
SCALE: 1:5
SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.
DRWN: S.HENG
CHKD:
DATE: 8-8-90
DRG No: D530

Fig 8.4





DEVELOPED SHAPE OF
50-100 DRAFT TUBE

DEVELOPED SHAPE OF
100mm FLOW STRAIGHTENER TUBE

DIMENSIONS

MACHINES	A	B	C	D	E	F	G	H	I
50x32-200	300	200	185	10	51	106	12	160	
65x40-200	300	200	185	10	51	106	12	160	
65x50-160	300	200	185	10	51	106	12	160	
80x65-160	600	300	285	10	77	156	12	200	
80x65-125	600	300	285	10	77	156	12	200	

MACHINES	I	J	K	L	M	N	O	P	Q
50x32-200	55	110	220	195	330	298	508	807	23.5
65x40-200	69	110	220	195	330	298			32.5
65x50-160	69	110	220	195	330	298			32.5
80x65-160	85	160	285	295	487	595			22
80x65-125	85	160	285	295	487	595			22

MACHINES		R
50x32-200	4 HOLES	18 EQUISPACED ON 114 PCD
65x40-200	"	" 127 PCD
65x50-160		" 127 PCD
80x65-160		" 146 PCD
80x65-125		" 146 PCD

MACHINES		S
50x32-200	4 HOLES	18 EQUISPACED ON 178 PCD
65x40-200		"
65x50-160		"
80x65-160	8 HOLES	18 EQUISPACED ON 146 PCD
80x65-125		"

ITEM	DESCRIPTION	QNTY

MATL: 2mm MILD STEEL SHEET

THIRD ANGLE PROJECTION | A1

TOLERANCES: unless otherwise stated

50mm, 65mm, 80mm
DRAFT TUBES

SCALE: 1 : 2

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S HENG

CHKD:

DATE: 5-9-90

DRG No: D520

Fig 8.5

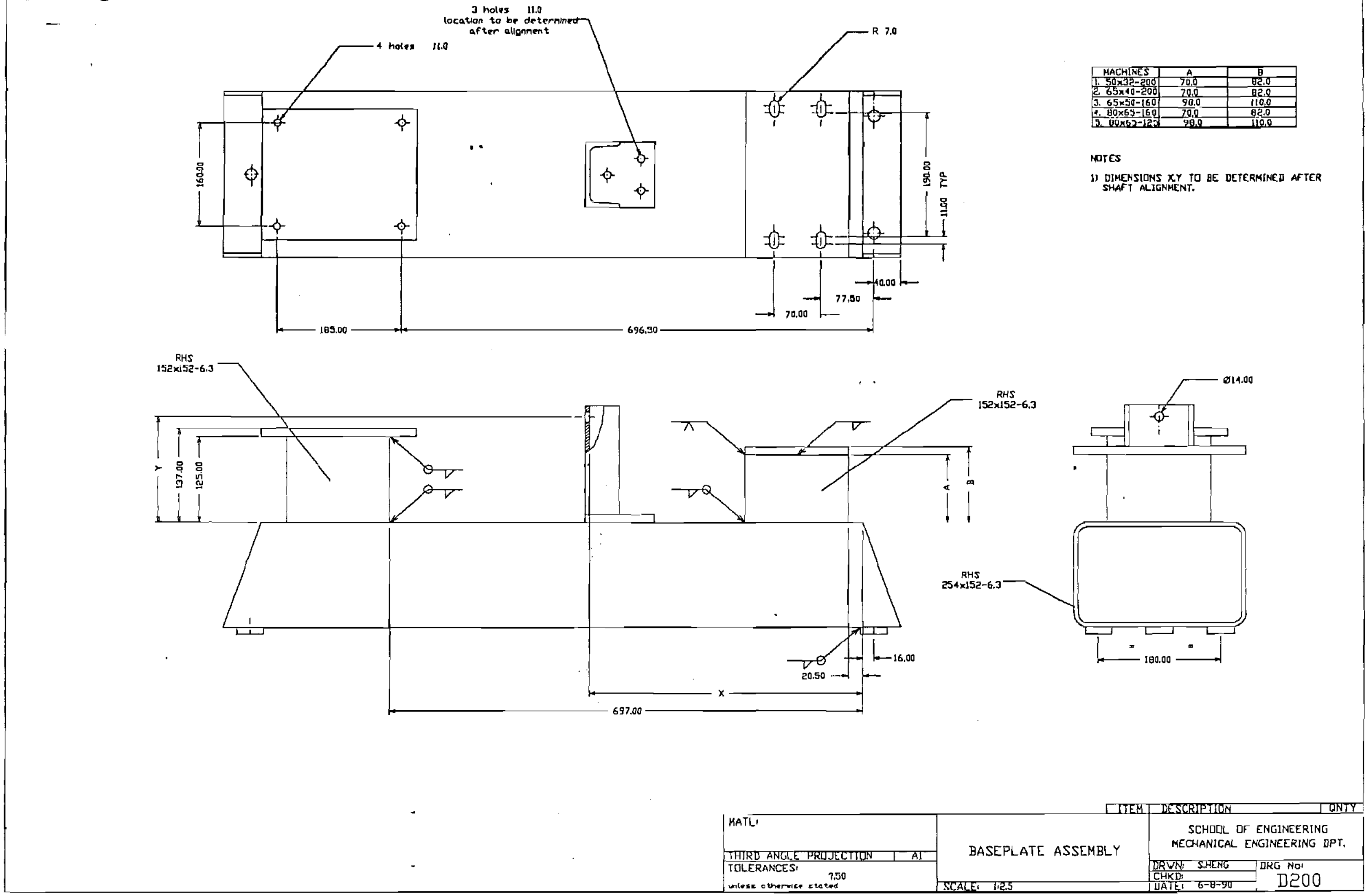
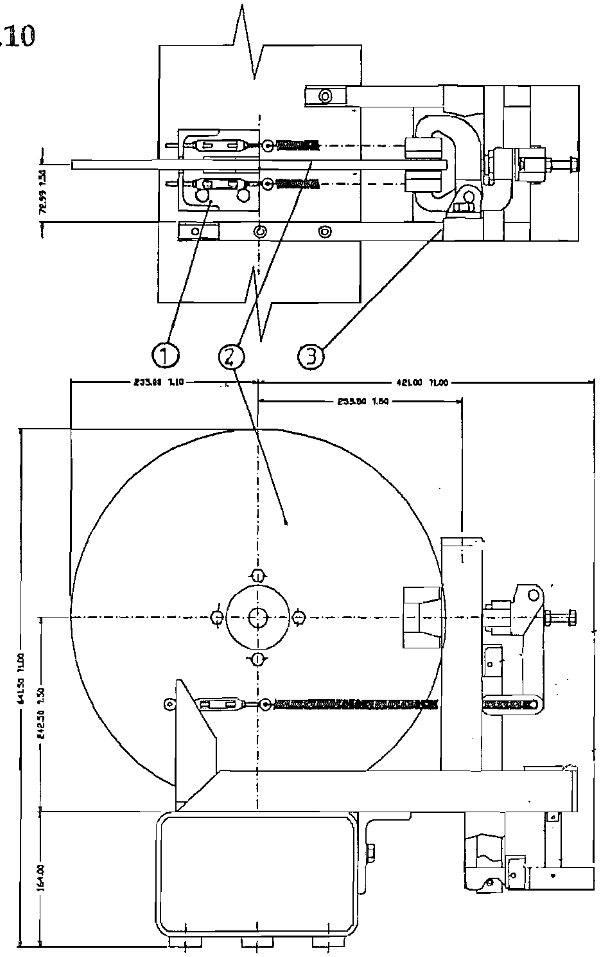
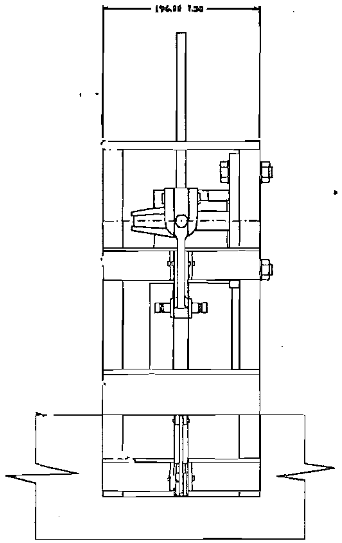

3 holes 11.0
location to be determined
after alignment
4 holes 11.0
R 7.0
160.00
190.00 TYP
11.00
10.00
77.50
70.00
185.00
696.50
RHS
152x152-6.3
Y
137.00
125.00
RHS
152x152-6.3
A
B
Ø14.00
RHS
254x152-6.3
16.00
20.50
X
697.00
180.00
MACHINES | A | B
1. 50x32-200 | 70.0 | 82.0
2. 65x40-200 | 70.0 | 82.0
3. 65x50-160 | 90.0 | 110.0
4. 80x65-160 | 70.0 | 82.0
5. 80x65-125 | 90.0 | 110.0
NOTES
1) DIMENSIONS X,Y TO BE DETERMINED AFTER SHAFT ALIGNMENT.
ITEM | DESCRIPTION | QNTY
MATL:
THIRD ANGLE PROJECTION | A1
TOLERANCES:
?.50
unless otherwise stated
BASEPLATE ASSEMBLY
SCALE: 1:2.5
SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.
DRWN: S.HENG
CHKD:
DATE: 6-8-90
DRG No:
D200

Fig 8.10



ITEMS	DESCRIPTION	DRG Nos.
1	Spring support	D320
2	Flywheel assembly	D110
3	Brake support assembly	D310



ITEM	DESCRIPTION	QNTY

MATL:

THIRD ANGLE PROJECTION A1

TOLERANCES:
unless otherwise stated

COMPLETE BRAKE RELEASE MECHANISM ASSEMBLY

SCALE: 1:2.5

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 18-8-90

DRG No: D300

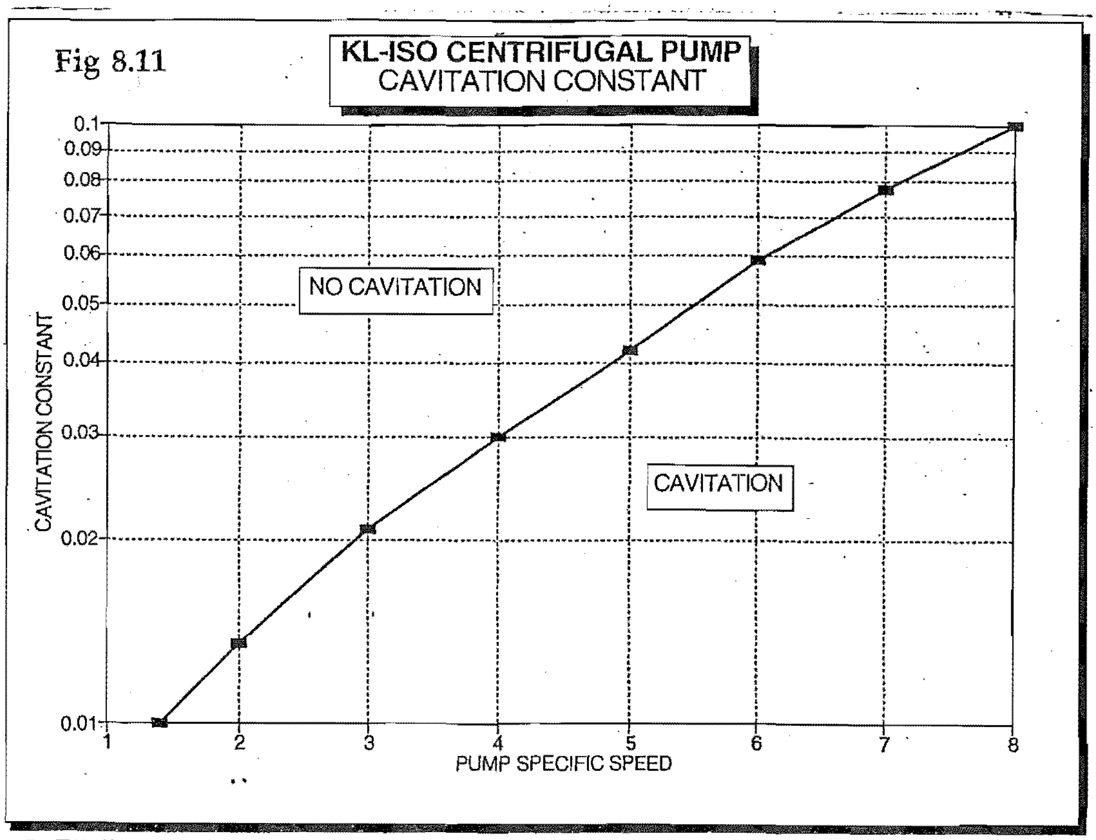


Fig 8.11
KL-ISO CENTRIFUGAL PUMP
CAVITATION CONSTANT
CAVITATION CONSTANT
0.1
0.09
0.08
0.07
0.06
0.05
0.04
0.03
0.02
0.01
NO CAVITATION
CAVITATION
1
2
3
4
5
6
7
8
PUMP SPECIFIC SPEED

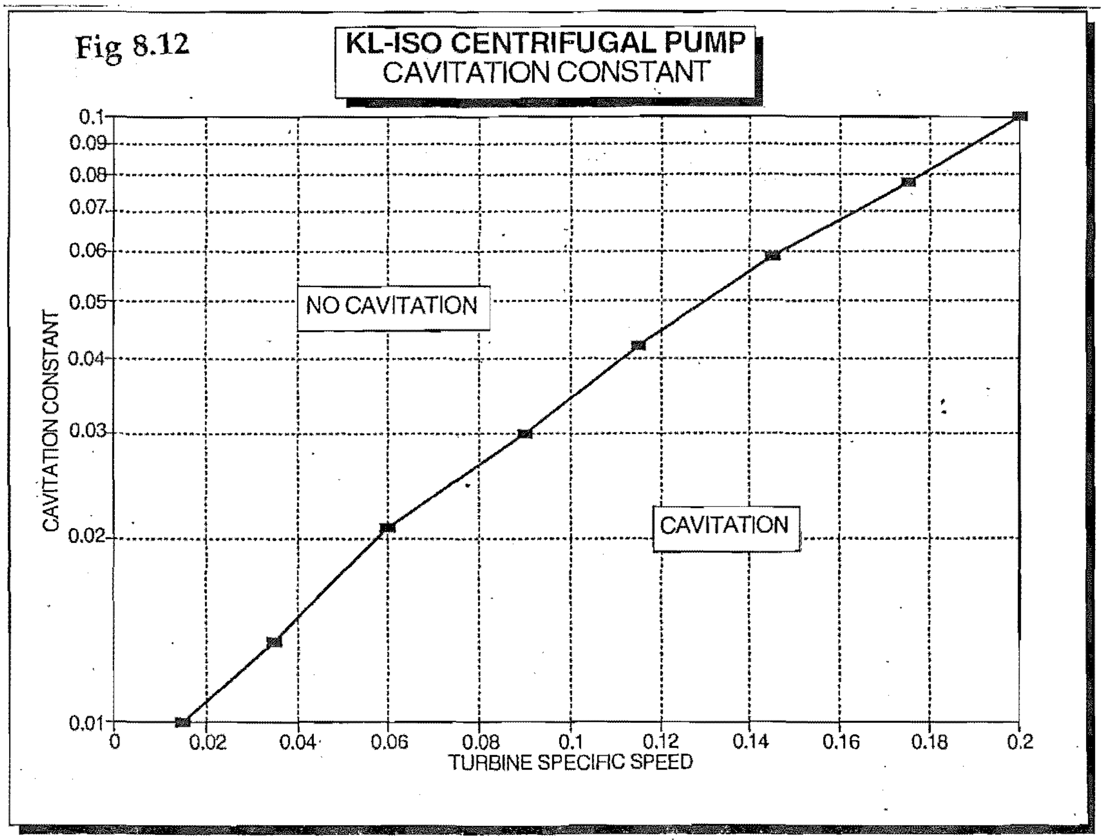


Fig 8.12
KL-ISO CENTRIFUGAL PUMP
CAVITATION CONSTANT
CAVITATION CONSTANT
0.1
0.09
0.08
0.07
0.06
0.05
0.04
0.03
0.02
0.01
NO CAVITATION
CAVITATION
0
0.02
0.04
0.06
0.08
0.1
0.12
0.14
0.16
0.18
0.2
TURBINE SPECIFIC SPEED

CHAPTER NINE

9 CONCLUSION AND RECOMMENDATION FOR FURTHER RESEARCH

At the outset, the task was to develop a viable microhydro generating set with an electrical capacity of 5 kW of single phase power at 230 v and 50 Hz, capable of supplying electricity at essentially urban standards of quality for use in isolated areas, assuming that a suitable water resource is available. Furthermore, the objects were to produce a design suitable for manufacturing and, at the same time, make provision for a range of different site conditions with a maximum of standardisation.

This was to be approached by using centrifugal pumps in the reverse mode of operation as water turbines and, by using a range of pumps from one source, to minimise the componentry. This was done and designs for five different site alternatives are the result.

The final arrangements have been determined by calculation and testing. The investigations produced conclusions on the relationships between the performance of the pumps in the pumping and turbining modes that are of more general significance, and provided experience of the effects of impeller trimming on turbining performance. They also include consideration of flywheel sizing in relation to governor performance.

Drawings and specification are provided together with an operating and maintenance manual.

From the point of view of further studies, it is tempting to consider using means of extending the range of options to lower heads with associated greater discharges. This is attractive because the higher specific speed machines show encouragingly good efficiencies when used as pumps or turbines.

Further studies might give consideration to fault diagnosis because attending to faults is a special problem with these installations where the operators have limited knowledge and the sites are remote from repair and maintenance resources. In this connection, consideration may well be given to system protection and load management being handled by micro-processors.

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APPENDIX 1

MANUFACTURING SPECIFICATION

MANUFACTURING SPECIFICATION

1 SCOPE

This document describes the work to be done and the materials to be used in the construction of a 5 kW microhydro electric generating set. It is to be read with the attached set of drawings numbered from D100 to D640 inclusive.

This document should be regarded as preliminary, and subject to finality only when the final manufacturing decisions are made.

2 GENERAL

The complete set of equipment ready for dispatch from the factory will comprise :-

1 a baseplate on which a turbine and generator are mounted
2 inlet and outlet pipework including valves and draft tube
3 an electronic load governor
4 switchboards complete with instruments
5 set of spares
6 Owner's installation, instruction and operation manual

3 MECHANICAL

3.1 TURBINE

The turbine takes the form of a KL-ISO Centrifugal pump operated in the reverse mode as a water turbine. Type and models are outlined in table A1.1. The unit will be fitted with one of five alternative pumps to suit the particular site conditions for full load operation producing 4.8 kW at 0.8 pf, in accordance with the following details:-

MACHINES	H (m)	Q (l/s)	η(%)
50x32-200	155	9	44
65x40-200[185]	85	13	56
65x50-160	64	14	70
80x65-160[142]	43	21	72
80x65-125	28	28	78

Table A1.1 KL-ISO turbine operating conditions

A packed gland is unsuitable for this application. A mechanical seal is required. The bearing support is to be modified in accordance with drawing number D200.

Impellers of the machines 65x50-200[185] and 80x65-160[142] are to be trimmed to the diameters specified in square brackets. This is necessary in order to achieve 6 kW shaft power at the best efficiency point.

The turbine is to be supplied complete with the pump manufacturer's operation and maintenance manual and complete list of manufacturers recommended spares.

3.2 COUPLING AND FLYWHEEL

The coupling is to be of rubber tyre flexible coupling type with a spacer which provides clearance of not less than 100 mm between stub shaft of turbine and stub shaft of the generator.

A Fenaflex type F50 rubber tyre coupling is to be used with SM16 spacer.

A disc flywheel of diameter 470 mm, 12 mm and inertia 0.41 kg/m^2 is to be mounted on the spacer coupling as shown on drawing number D110. The flywheel is to be statically balanced using knife edge pivot before being fitted to the coupling.

Ensure proper alignment of the turbine-generator assembly. Bear in mind that a flexible coupling will permit end movement in the axial direction but will not avoid the need for accurate alignment between the turbine and generator shaft. Dial indicators should be used to check angular alignment and parallel alignment.

3.3 BRAKES AND RELEASE MECHANISM

The brake and spring release mechanism is shown on drawing number D300. The brake is to be Twiflex MSF mechanical calliper or similar type.

Two tension springs are used to release the brake lever arm. These springs are to be of stainless type with initial length 180 mm and spring constant 1.1 N/mm. The pre-tension is to be adjustable.

3.4 BASEPLATE

The baseplate is to be a single RHS, on which the turbine, generator and the brake assembly are mounted. The baseplate should be constructed to provide accurate turbine-generator shaft align. Doweling of the turbine base, turbine bearing support and the generator should be done once the alignment is established.

3.5 PIPEWORK

All pipe and fittings shall comply with the following documents:-

1	BS 4772	Pipe and fittings
2	BS 4504	Flange and bolting for pipes, valves and fitting, Metric series (Part 1)

The pipework is to be fitted with two valves for inlet water supply and flushing of pipeline. The valves are to be geared butterfly valves with a one piece cast stainless steel butterfly unit. Catalogue number F990 CEE3 of the Keystone range or equivalent standard valves suitable for installing between flanges.

All valves shall be provided with geared handwheel.

A pressure gauge is to be fitted upstream of the two valves. The gauge is to have a span of 0 to 1500 kPa.

4 ELECTRICAL

All electrical equipment shall comply with the New Zealand Electrical wiring regulations, and the Manufacturer's instruction.

4.1 GENERATOR

The generator is to be a 6 kVA, 230 v, single phase, 50 Hz, 2-pole, brushless generator with drip-proof enclosure. It must be suitable for continuous operation at 3000 rpm at full load. A Markon type B21 or similar is to be used. It is to fitted with ball or roller bearings that can be replaced in the field. The wiring is to be suitable for terminating into a 3 pin 30 Amps socket and mounting box.

The generator is to be supplied complete with operation and maintenance manual, list of parts and supplied with a set of the manufacturer's recommended spares.

4.2 LOAD GOVERNING

A method of load governing is required to control the frequency to 50 Hz. An electronic load governor of the following specification is to be used:-

1. Rated power not less than 6 kW
2. Operate on 230 volt, 50 Hz power supply source
3. Frequency droop not more than 3%
4. To be supplied with matching dummy load banks

A governor manufactured by Delphi Industries of Auckland rated 12 kW which operates on 230 v, 50 Hz is to be used. Similar governors may be used if manufactured to similar specifications.

4.3 MALFUNCTION PROTECTION EQUIPMENT

a) Under/over voltage protection

Under voltage trip shall occur when line voltage drops below 180 Volts. Over voltage trip shall occur when the line voltage exceeds 260 volts. If voltage goes out of the set range, the brake is required to be stall the plant.

b) Under/over frequency (speed) protection

Under frequency trip shall occur when the system frequency drops below 45 Hz. Over frequency trip shall occur when the system frequency exceeds 55 Hz. If frequency goes out of the set range, the brake is required to stall the plant.

c) Over current protection

Over current trip shall occur when the current reaches a set maximum of 30 Amps. It shall be so arranged to be safe of any nuisance tripping from normal system transient. If over current occurs, the brake is required to stall the plant.

4.4 OVERLOAD WARNING SYSTEM

Monitoring of overloading is to be done by sensing frequency. Overload warning conditions exist if the frequency drops to less than 48 Hz. An audio alarm and a timer shall be energised upon overload. The timer shall be user adjustable from 5 to 20 seconds. Disconnection of the consumer shall be initiated if overload continues for more than the set time period. If load is reduced within the time limit, the timer and the alarms are to be reset.

4.5 MONITORING AND DISPLAY INSTRUMENTS

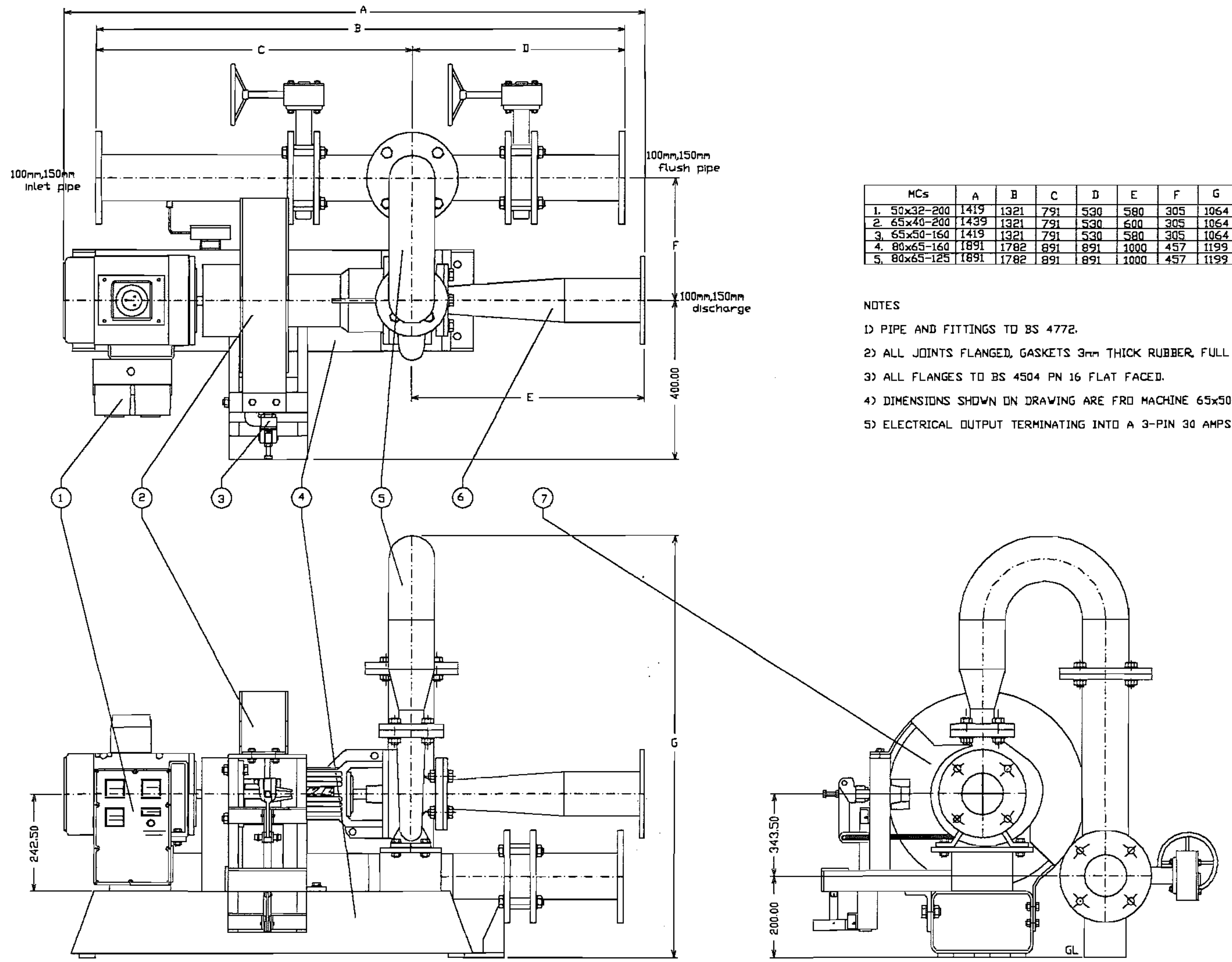
The following instruments shall be provided for the malfunction protection equipment:-

1 An Ammeter with span of 0 to 30 amps
2 A voltmeter with span of 0 to 300 volts
3 A frequency meter with span of 40 to 60 Hz
4 A run-hour meter with span up to 100,000 hours
5 A weather-proof storage cabinet

APPENDIX 2

COMPLETE SET OF DRAWINGS

1	GENERAL ASSEMBLY	D100-D110
2	BASEPLATE ASSEMBLY	D200-D210
3	BRAKE RELEASE MECHANISM ASSEMBLY	D300-D375
4	FLYWHEEL GUARD	D400-D410
5	PIPEWORK DESIGN	D500-D560
6	ELECTRICAL	D600-D640



MCs	A	B	C	D	E	F	G	PIPE
1. 50x32-200	1419	1321	791	530	580	305	1064	100
2. 65x40-200	1439	1321	791	530	600	305	1064	100
3. 65x50-160	1419	1321	791	530	580	305	1064	100
4. 80x65-160	1891	1782	891	891	1000	457	1199	150
5. 80x65-125	1891	1782	891	891	1000	457	1199	150

NOTES

1) PIPE AND FITTINGS TO BS 4772.

2) ALL JOINTS FLANGED, GASKETS 3mm THICK RUBBER, FULL FACED.

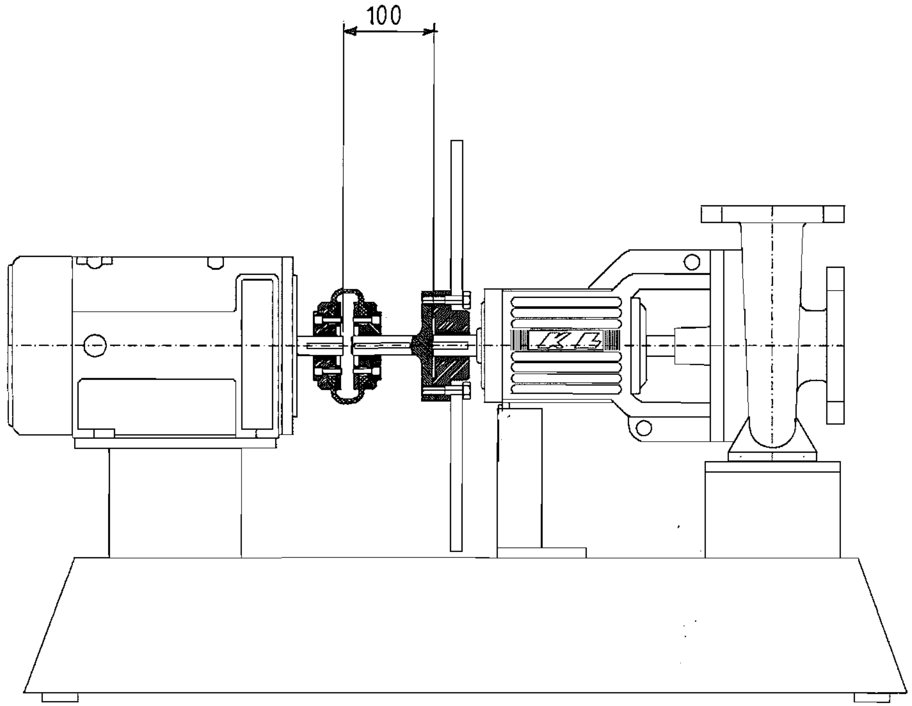
3) ALL FLANGES TO BS 4504 PN 16 FLAT FACED.

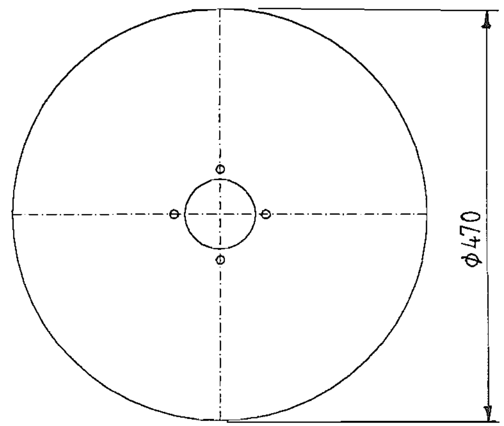
4) DIMENSIONS SHOWN ON DRAWING ARE FRO MACHINE 65x50-160.

5) ELECTRICAL OUTPUT TERMINATING INTO A 3-PIN 30 AMPS SOCKET

ITEM	DESCRIPTION	DRG No
7	Coupling assembly	D110
6	Draft tube	D520
5	Pipework	D530
4	Baseplate	D200
3	Brake assembly	D300
2	Coupling guard	D400
1	Electrical protection system	D600

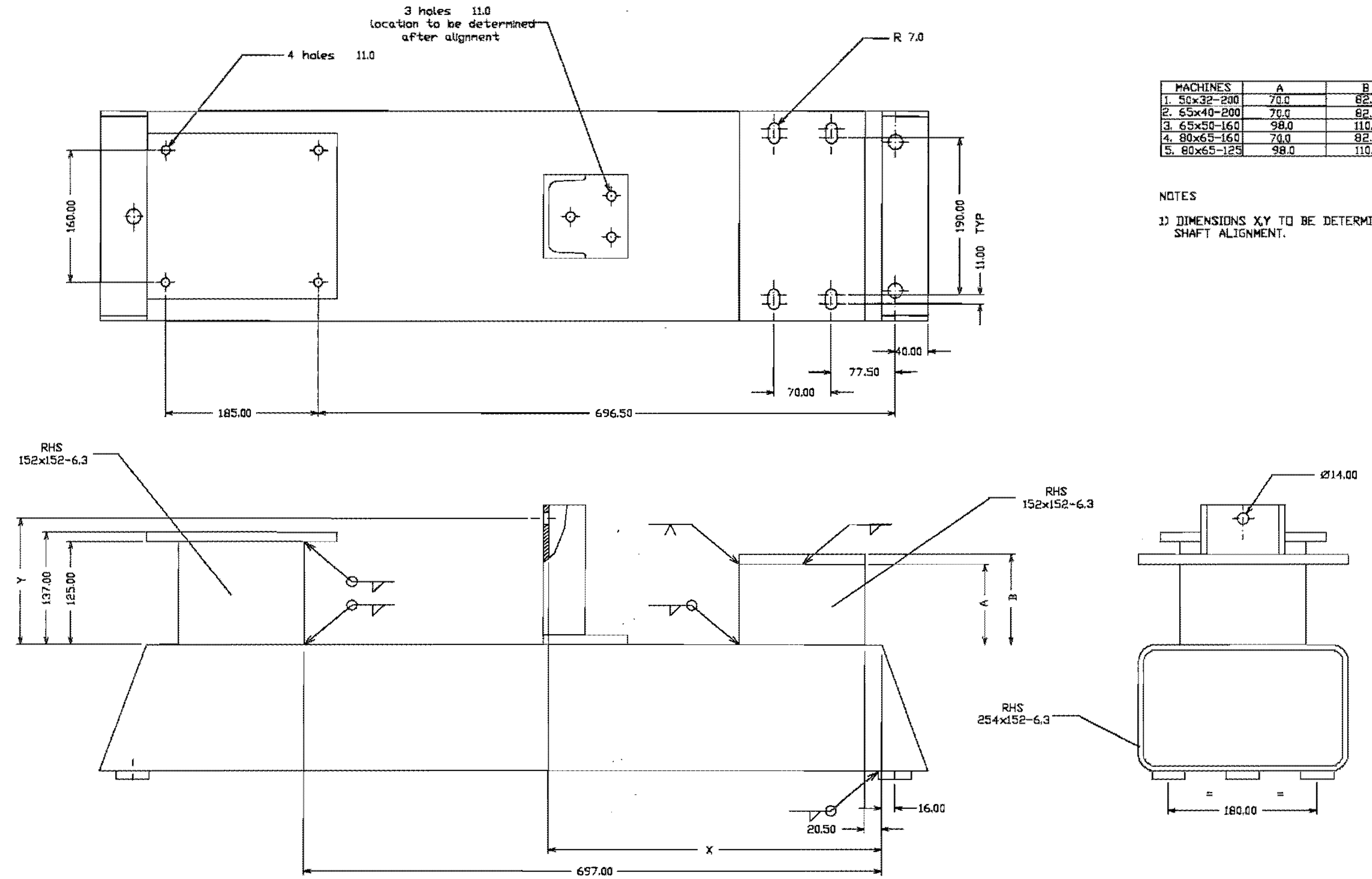
MATL:	5kW MICROHYDRO GENERATING SET	SCHOOL OF ENGINEERING MECHANICAL ENGINEERING DPT.
THIRD ANGLE PROJECTION A1		DRWN: S.HENG
TOLERANCES: unless otherwise stated		CHKD:
	SCALE: 1:5	DATE: 10-8-90
		DRG No: D100





ITEM	DESCRIPTION	QNTY

MATL:	COUPLING AND FLYWHEEL ASSEMBLY	SCHOOL OF ENGINEERING MECHANICAL ENGINEERING DPT.
THIRD ANGLE PROJECTION A1		DRWN: S. HENG
TOLERANCES: unless otherwise stated	SCALE: 1:2.5	CHKD: DATE: 10-8-90
		DRG No: D110



MACHINES	A	B
1. 50x32-200	70.0	82.0
2. 65x40-200	70.0	82.0
3. 65x50-160	98.0	110.0
4. 80x65-160	70.0	82.0
5. 80x65-125	98.0	110.0

NOTES

1) DIMENSIONS X,Y TO BE DETERMINED AFTER SHAFT ALIGNMENT.

ITEM	DESCRIPTION	QNTY

MATL:

THIRD ANGLE PROJECTION | A1

TOLERANCES: ?.50 unless otherwise stated

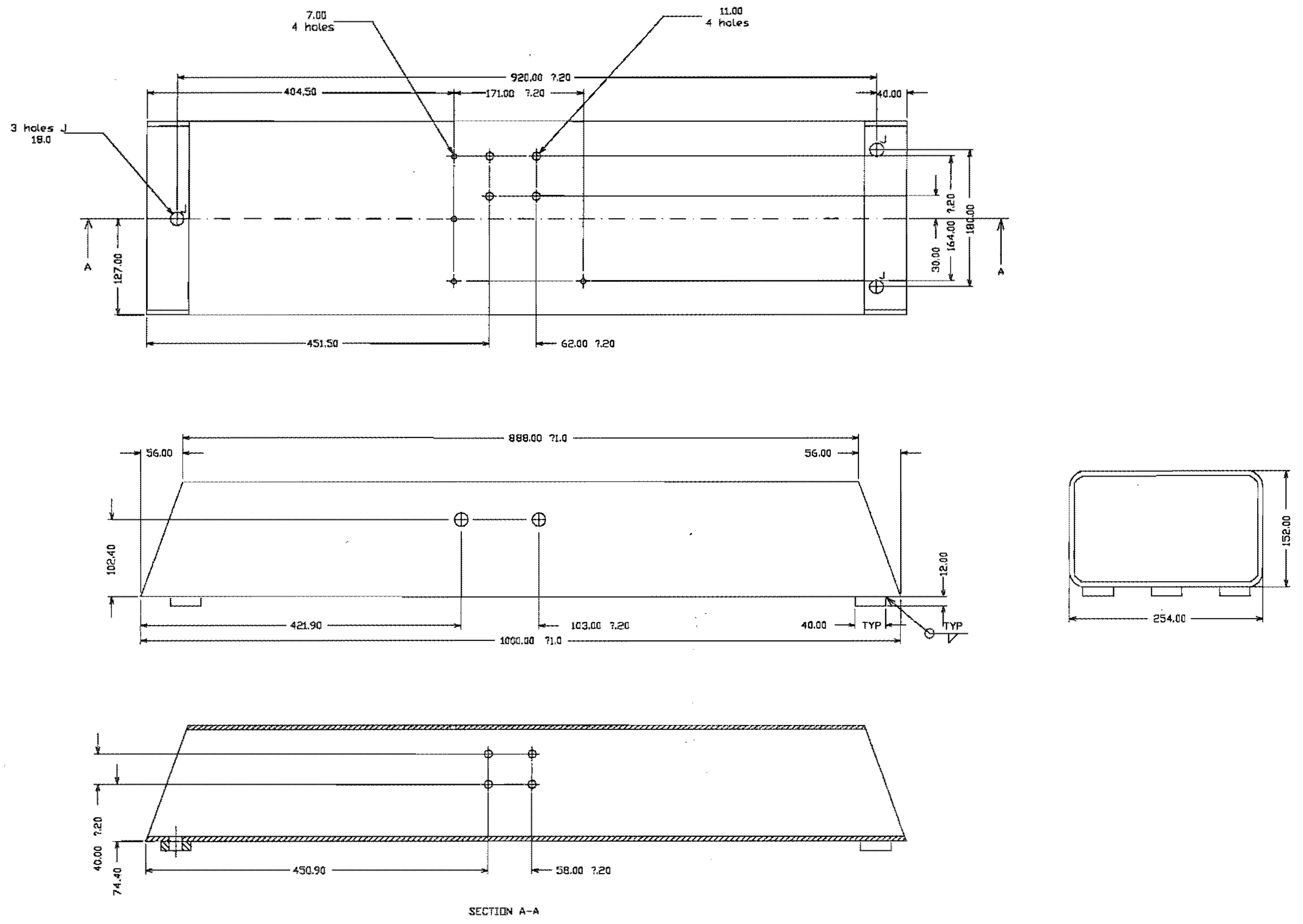
BASEPLATE ASSEMBLY

SCALE: 1:2.5

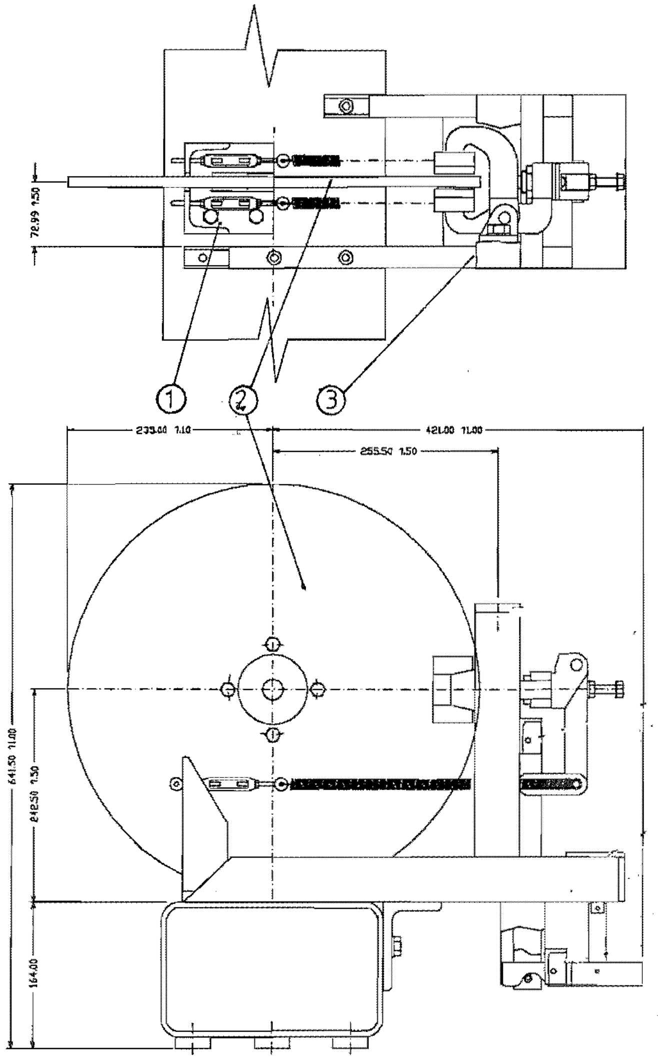
SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 6-8-90

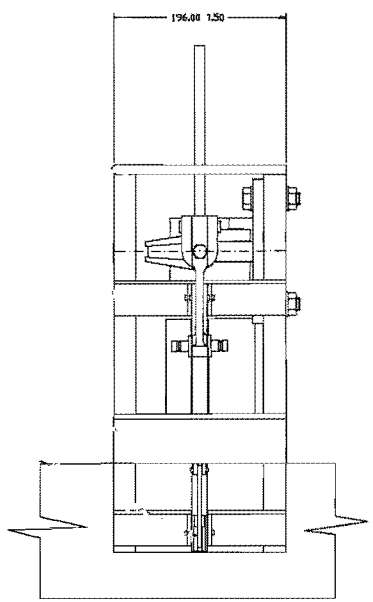
DRG No: D200



		ITEM	DESCRIPTION / QNTY
MATL: MS 254x152-6.3 RHS	RHS BASE PLATE	SCHOOL OF ENGINEERING MECHANICAL ENGINEERING DPT.	
THIRD ANGLE PROJECTION / A1		DRWN: S.HENG	DRG No: D210
TOLERANCES: ?.50 unless otherwise stated	SCALE: 1:2.5	CHKD:	
		DATE: 2-8-90	



ITEMS	DESCRIPTION	DRG Nos.
1	Spring support	D320
2	Flywheel assembly	D110
3	Brake support assembly	D310



ITEM	DESCRIPTION	QNTY

MATL:

THIRD ANGLE PROJECTION | A1

TOLERANCES:
unless otherwise stated

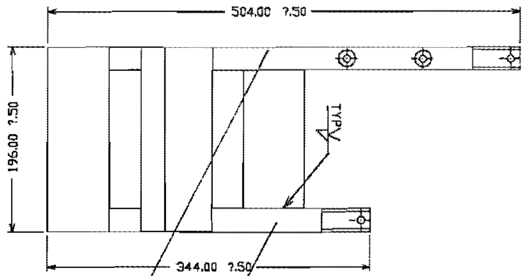
COMPLETE BRAKE RELEASE
MECHANISM ASSEMBLY

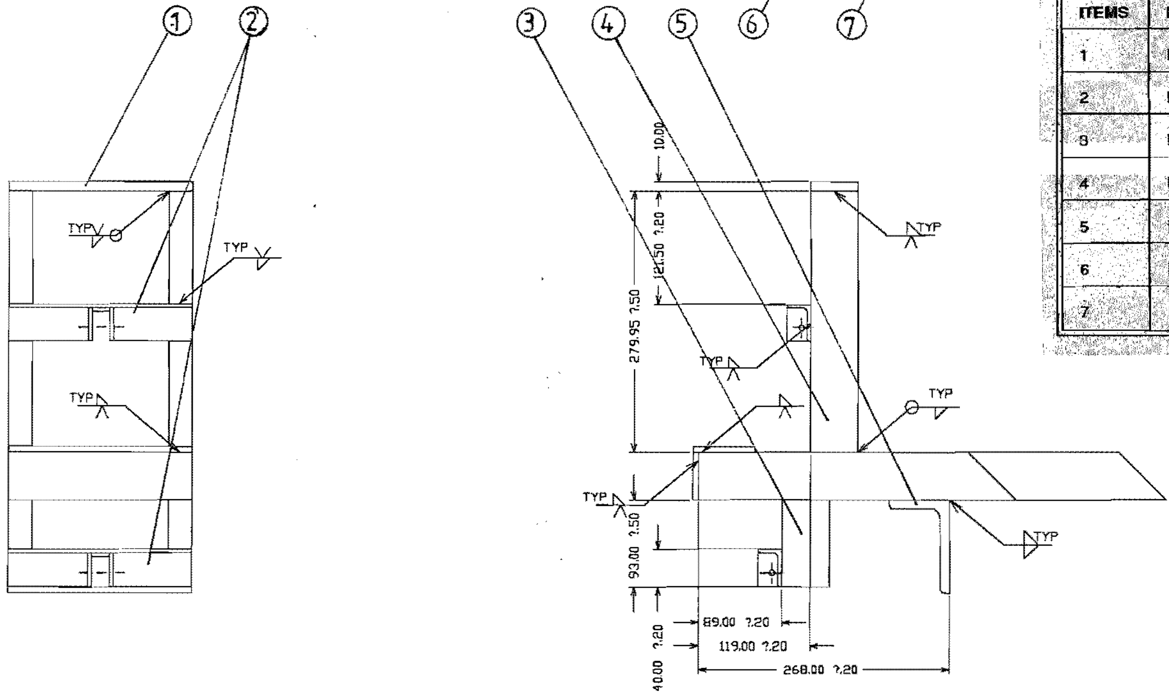
SCALE: 1:2.5

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 18-8-90

DRG No: D300





ITEMS	DESCRIPTION	DRG Nos
1	Part #3	D335
2	Part #1	D325
3	Part #x	D355
4	Part #x	D335
5	Part #2	D330
6	Part #4	D340
7	Part #5	D345

ITEM	DESCRIPTION	QNTY

MATL:

THIRD ANGLE PROJECTION | A1

TOLERANCES:
unless otherwise stated

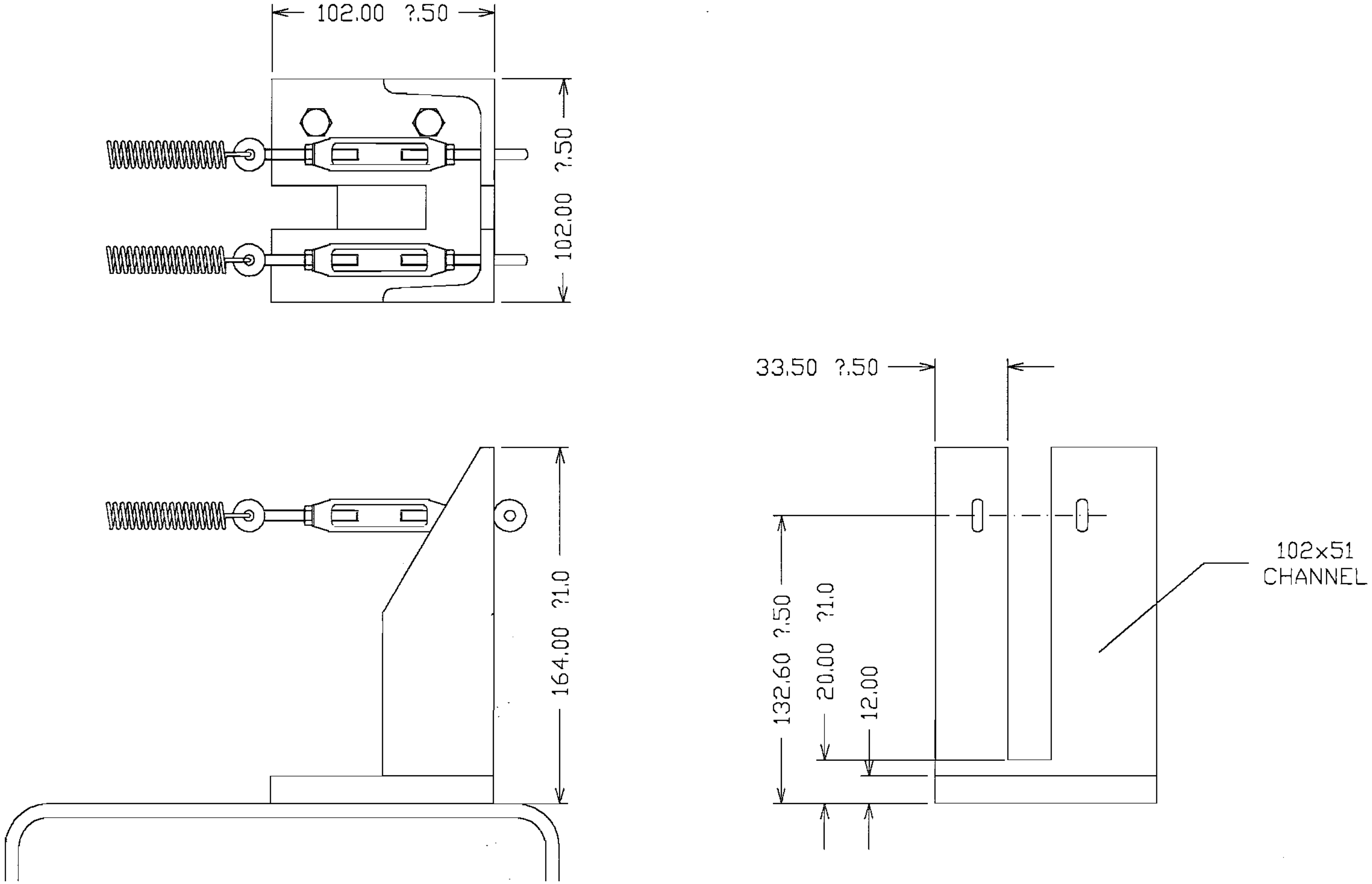
BRAKE SUPPORT ASSY

SCALE: 1 : 2.5

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 6-8-90

DRG No: D310



ITEM	DESCRIPTION	QNTY

MATL:
MILD STEEL

THIRD ANGLE PROJECTION | A3

TOLERANCES:
unless otherwise stated

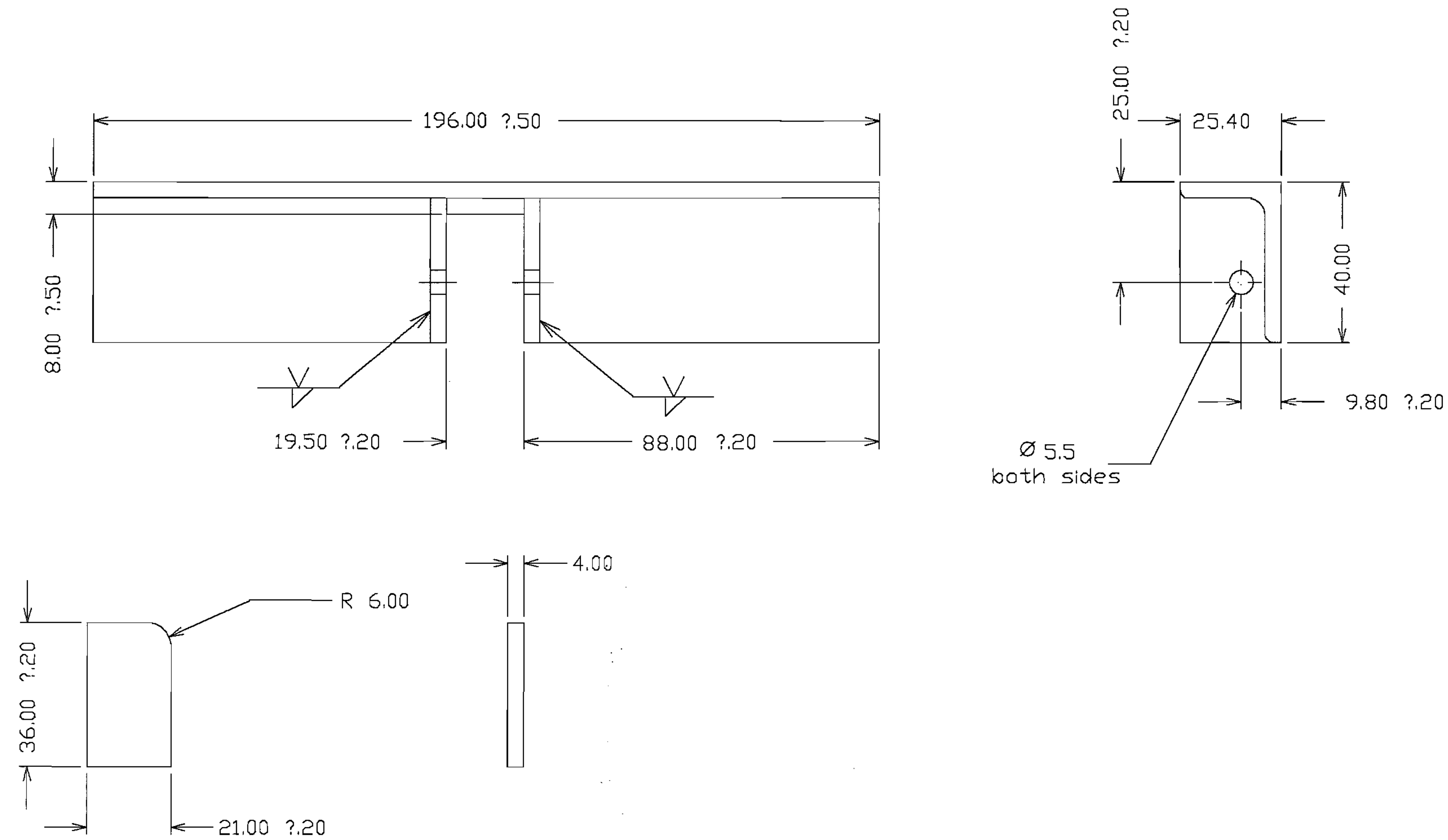
SPRING SUPPORT

SCALE: 1:2

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 14-8-90

DRG No: D320



ITEM	DESCRIPTION	QNTY

MATL:
mild steel

THIRD ANGLE PROJECTION | A3

TOLERANCES:
unless otherwise stated

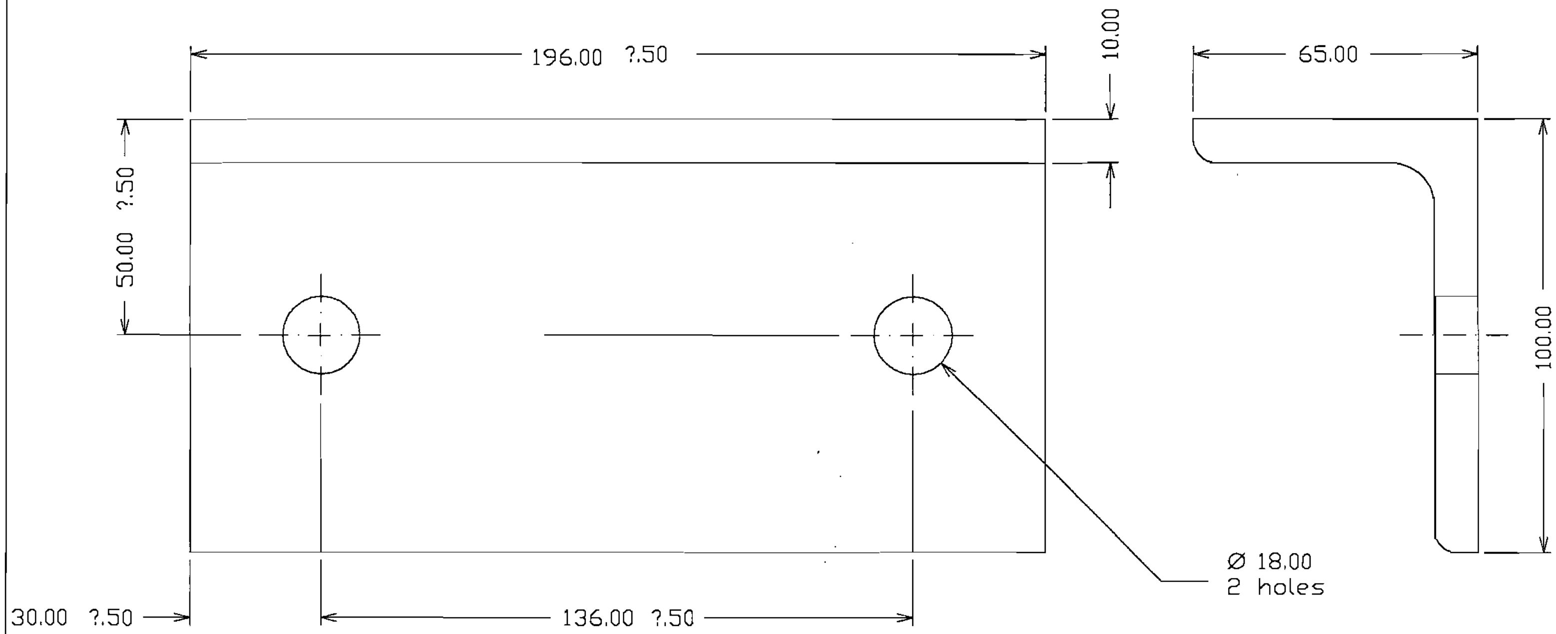
BRAKE SUPPORT PART #1

SCALE: 1:1

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 15-8-90

DRG No:
D325



ITEM	DESCRIPTION	QNTY

MATL:
MS 100x65 angle

THIRD ANGLE PROJECTION | A3

TOLERANCES:
unless otherwise stated

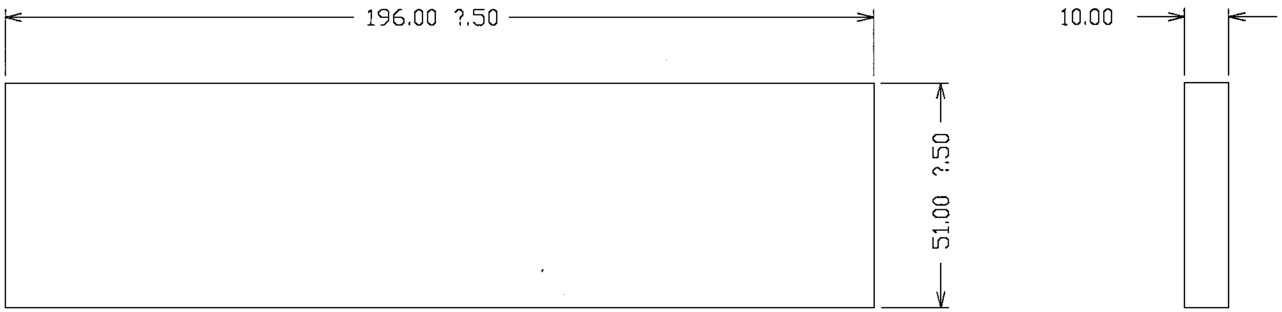
BRAKE SUPPORT PART #2

SCALE: 1:1

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 15-8-90

DRG No:
D330



ITEM	DESCRIPTION	QNTY

MATL:
MILD STEEL

THIRD ANGLE PROJECTION | A3

TOLERANCES:
unless otherwise stated

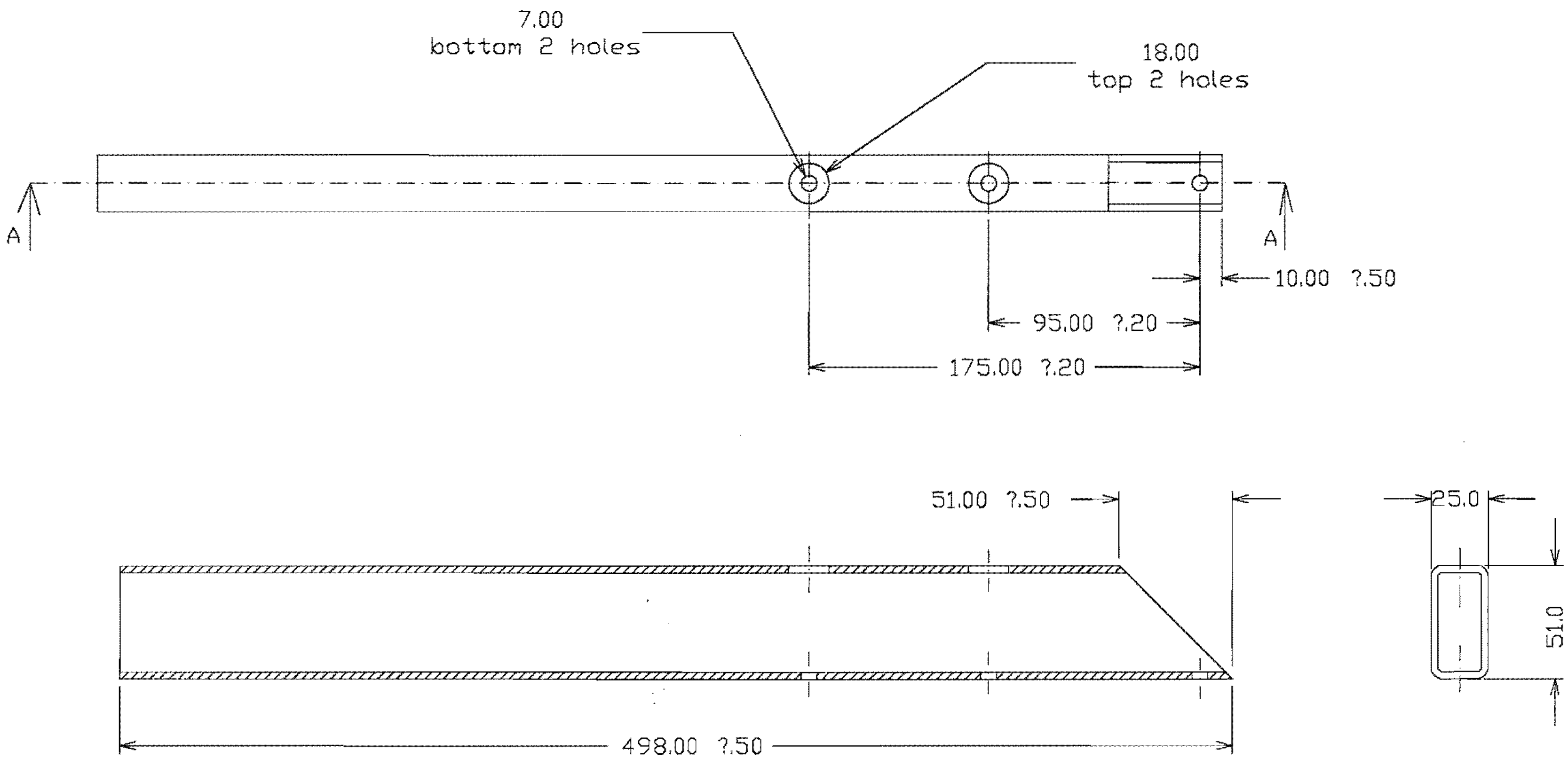
BRAKE SUPPORT PART #3

SCALE: 1:1

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 15-8-90

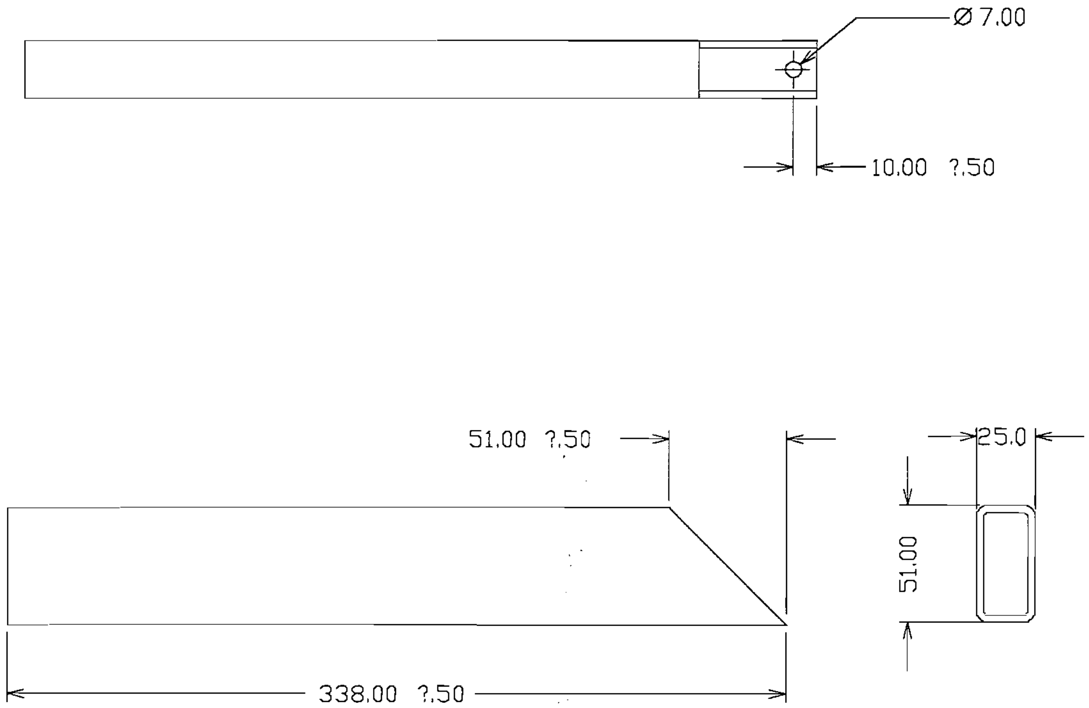
DRG No:
D335



SECTION A-A

ITEM	DESCRIPTION	QNTY

MATL: 51x25 MS RHS	BRAKE SUPPORT PART #4	SCHOOL OF ENGINEERING MECHANICAL ENGINEERING DPT.
THIRD ANGLE PROJECTION — A3		DRWN: S.HENG
TOLERANCES:		CHKD:
unless otherwise stated	SCALE: 1:2	DATE: 2-8-90
		DRG No: D340



ITEM	DESCRIPTION	QNTY

MATL:
51x25 MS RHS

THIRD ANGLE PROJECTION | A3

TOLERANCES:
unless otherwise stated

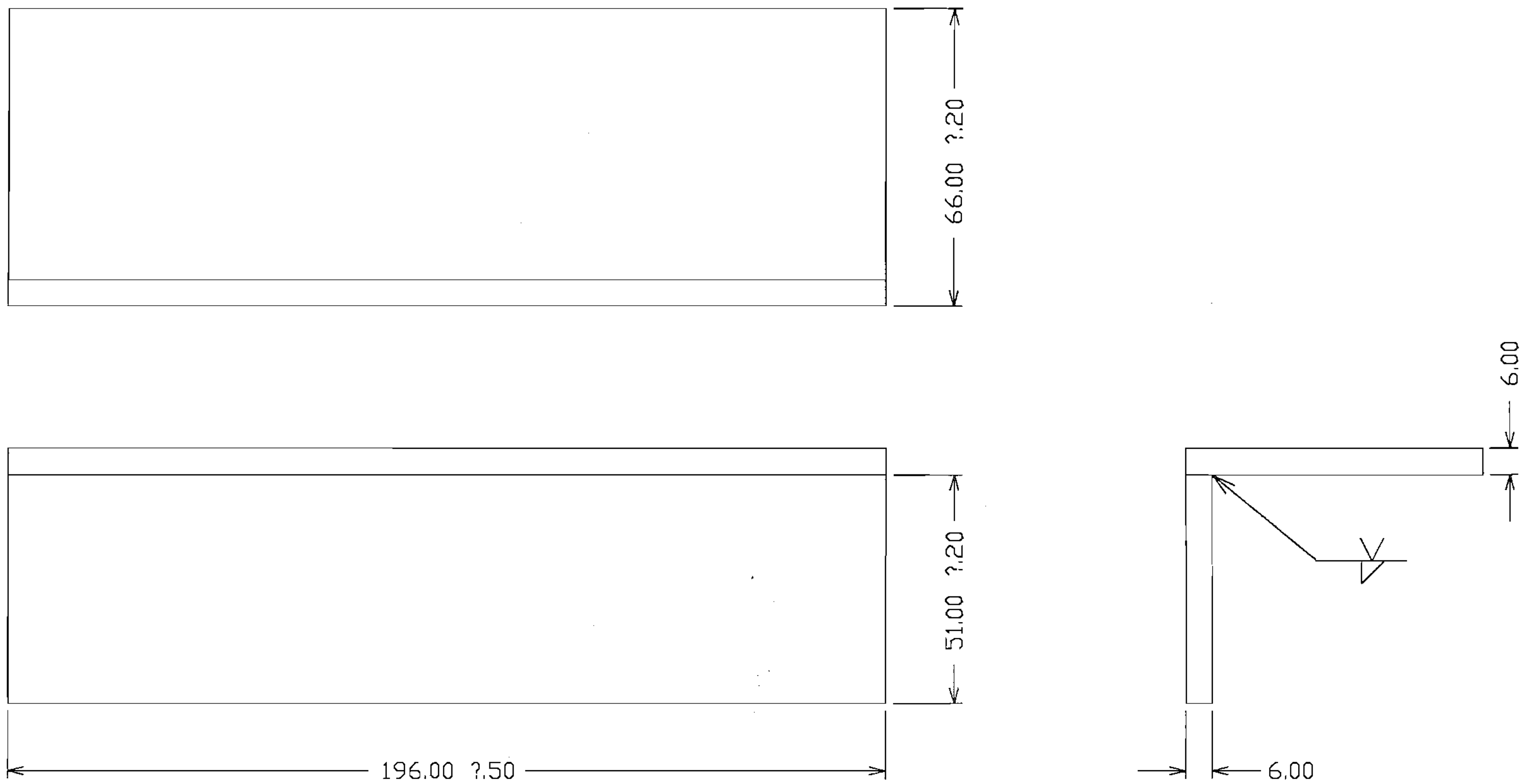
BRAKE SUPPORT FRAME
PART #5

SCALE: 1:2

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 3-9-90

DRG No:
D345



ITEM	DESCRIPTION	QNTY

MATL:
6mm MS plate

THIRD ANGLE PROJECTION | A3

TOLERANCES:
unless otherwise stated

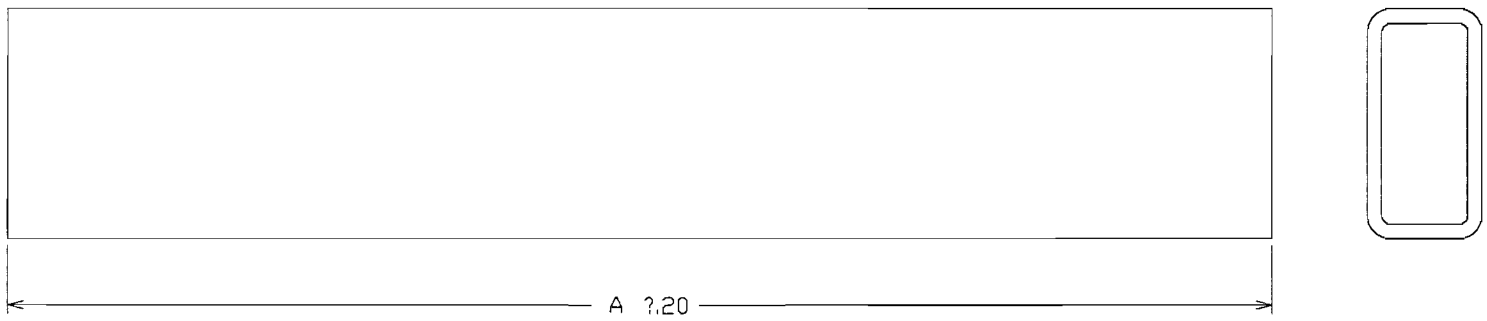
BRAKE SUPPORT PART #6

SCALE: 1:1

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 15-8-90

DRG No:
D350



PART #X	A	NUB RQRD
7	280	2
8	93	2

ITEM	DESCRIPTION	QNTY

MATL:

51x25-3.2 RHS

THIRD ANGLE PROJECTION | A3

TOLERANCES:

unless otherwise stated

BRAKE SUPPORT
PART #X

SCALE: 1:1

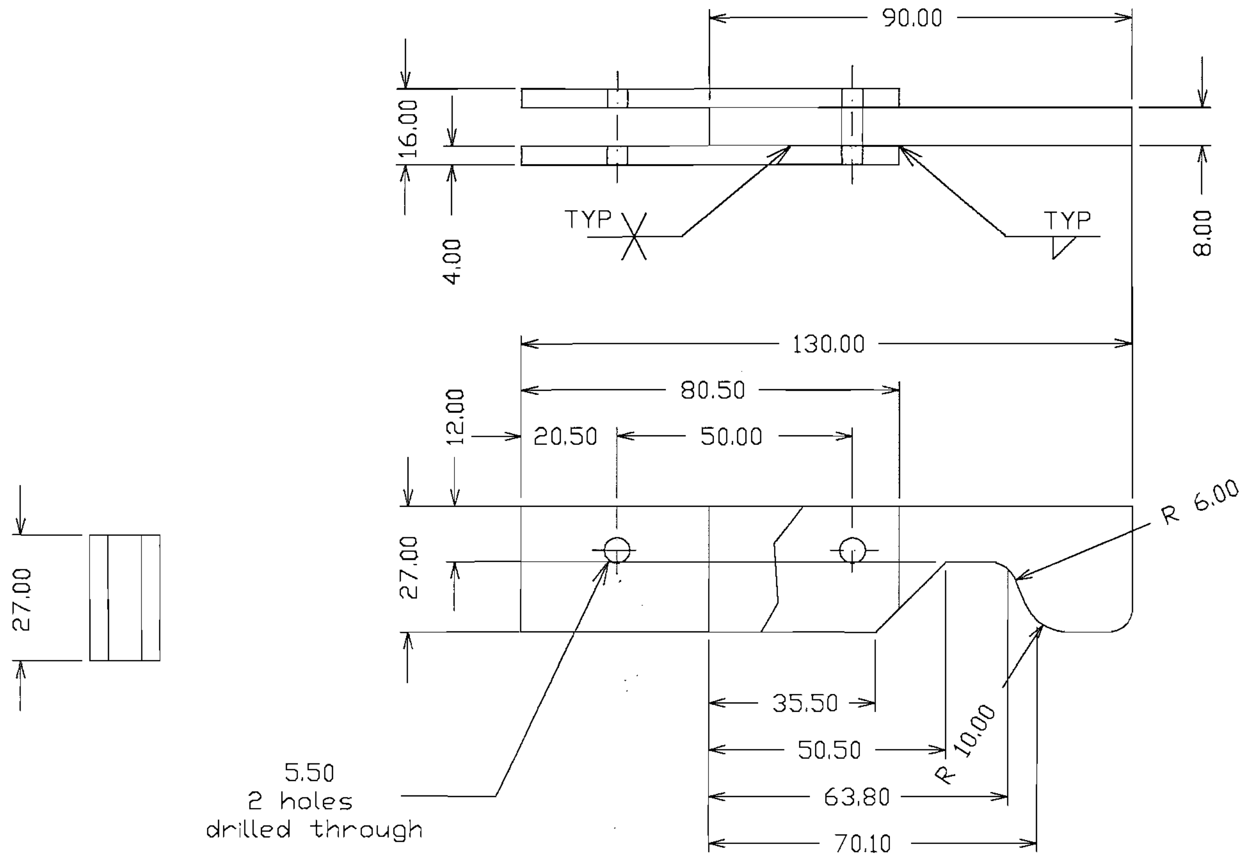
SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG

CHKD:

DATE: 15-8-90

DRG No: D355



ITEM	DESCRIPTION	QNTY

MATL:
MS

THIRD ANGLE PROJECTION | A3

TOLERANCES:
?.20
unless otherwise stated

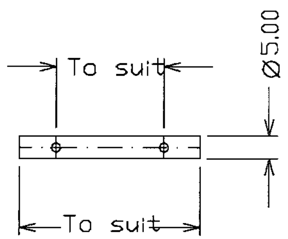
BRAKE RELEASE LATCH
PART #1

SCALE: 1:1

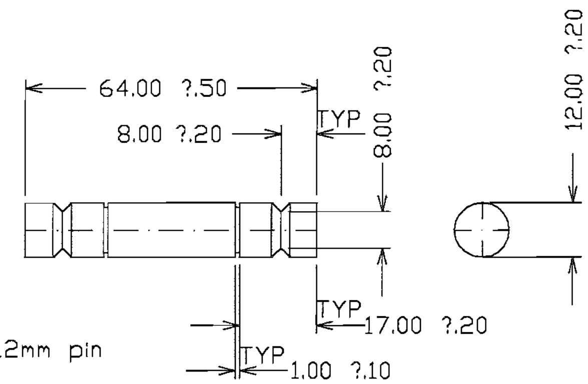
SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 9-8-90

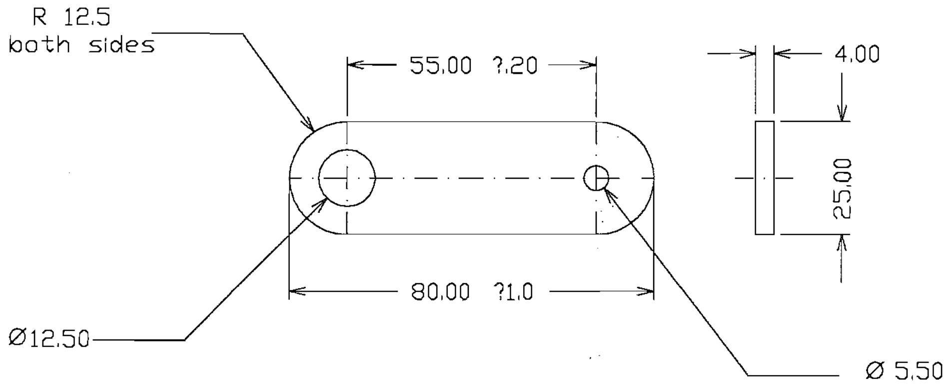
DRG No:
D360



5mm pin



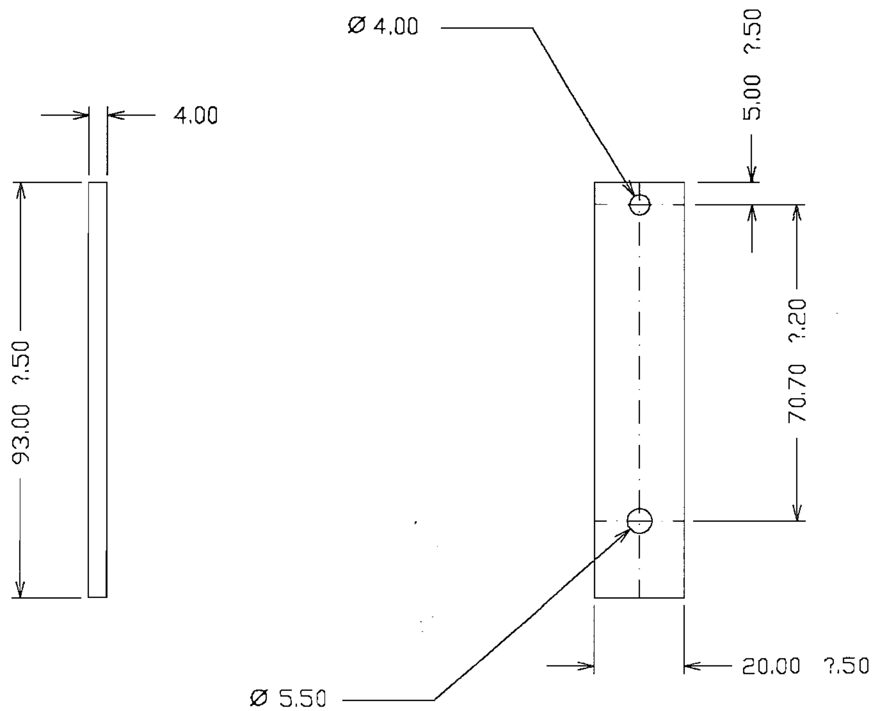
12mm pin



No. REQUIRED=2

ITEM	DESCRIPTION	QNTY

MATL: ms	BRAKE LEVER PINS AND CONNECTORS	SCHOOL OF ENGINEERING MECHANICAL ENGINEERING DPT.
THIRD ANGLE PROJECTION — A3		DRWN: S.Heng
TOLERANCES: unless otherwise stated		CHKD:
	SCALE: 1:1	DATE: 18-8-90
		DRG No: D365



ITEM	DESCRIPTION	QNTY

MATL:

MS

THIRD ANGLE PROJECTION | A3

TOLERANCES:

unless otherwise stated

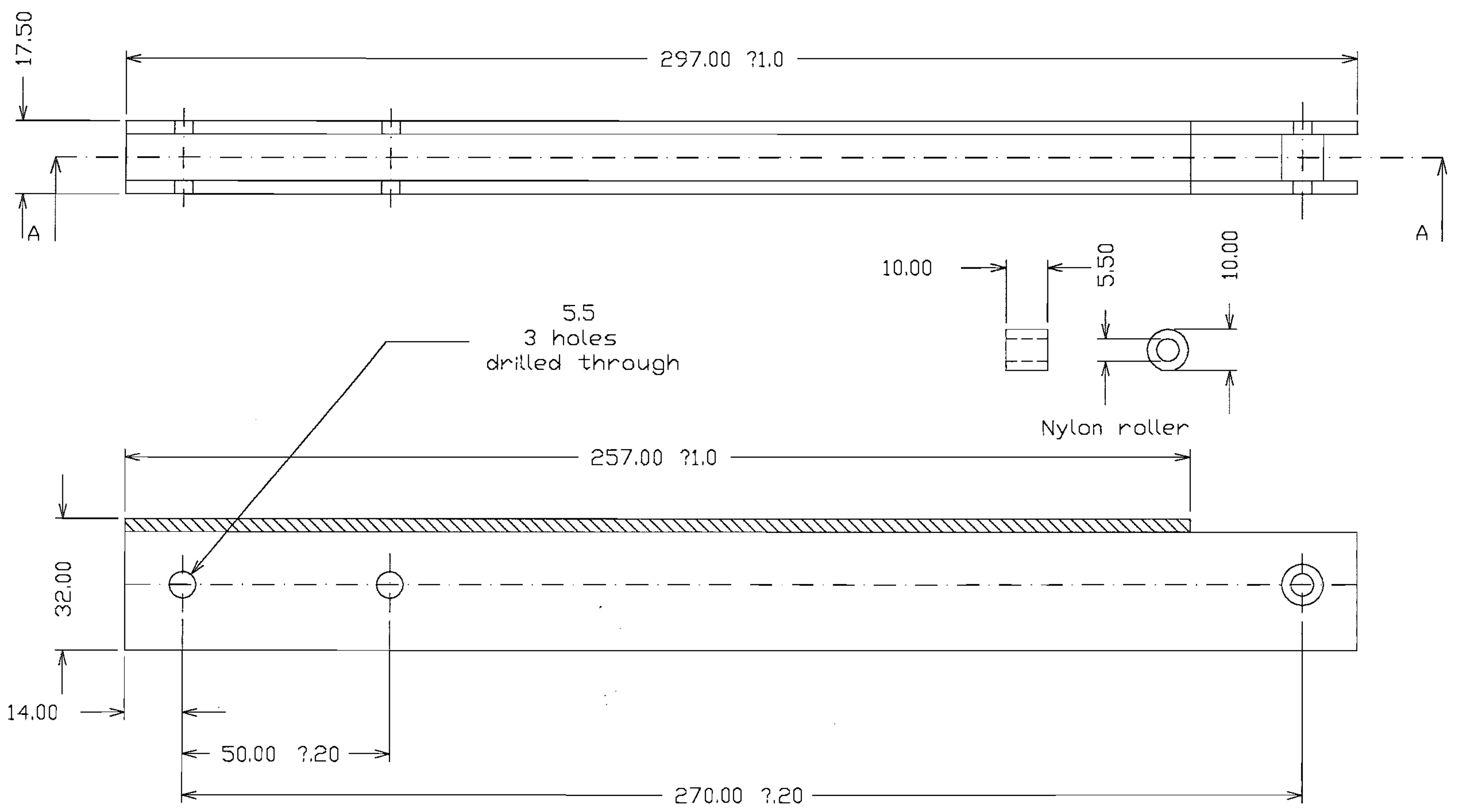
BRAKE RELEASE LATCH
PART #2

SCALE: 1:1

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 18-8-90

DRG No:
D370



SECTION A-A

ITEM	DESCRIPTION	QNTY

MATL: 32x17.5-3.2 ALUMINIUN SECTION

THIRD ANGLE PROJECTION — A3

TOLERANCES: ?.50 unless otherwise stated

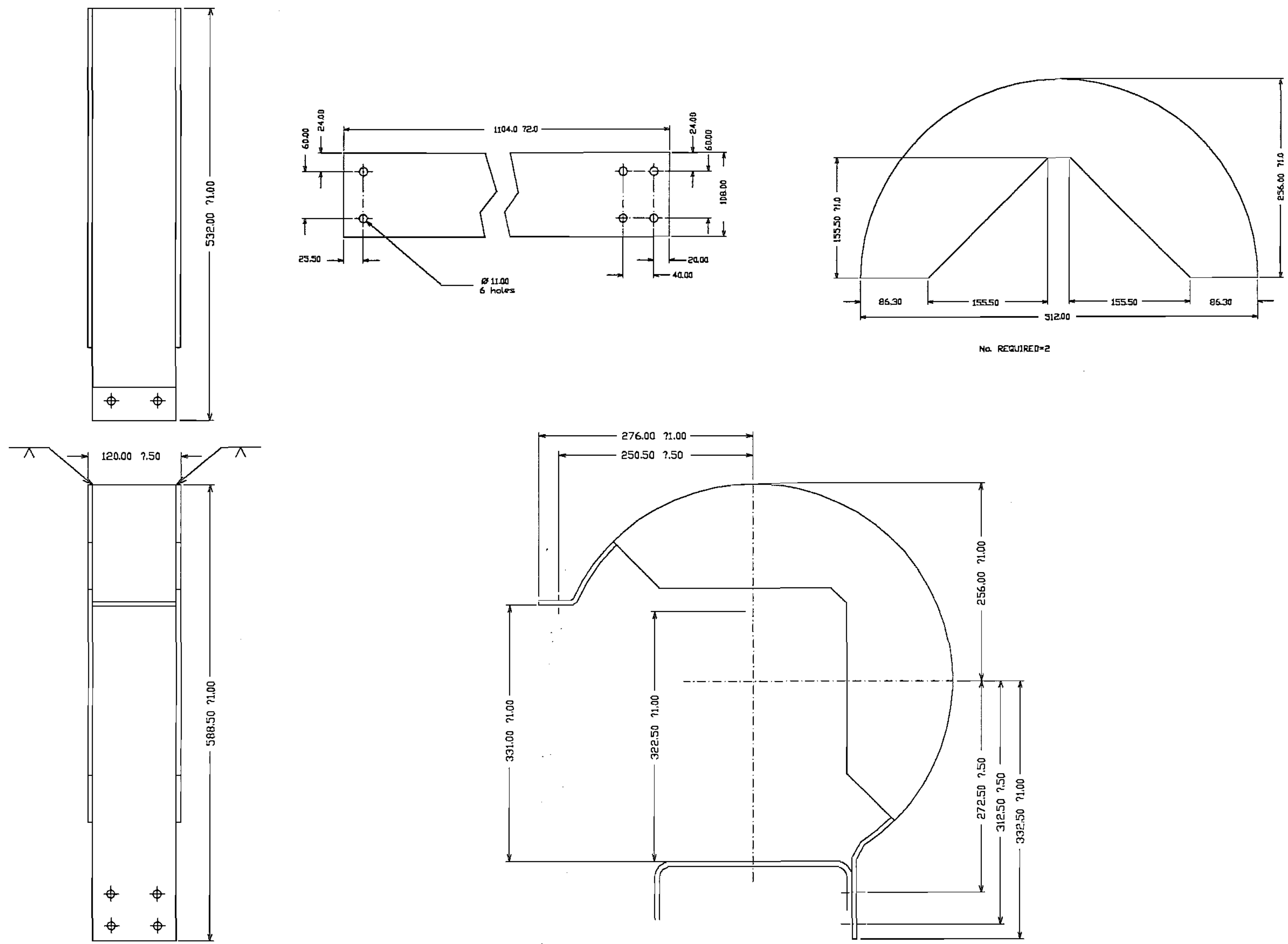
BRAKE CONTROL LEVER

SCALE: 1:1

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 3-8-90

DRG No: D375



ITEM	DESCRIPTION	QNTY

MATL: 6mm MS plate

THIRD ANGLE PROJECTION A1

TOLERANCES:
unless otherwise stated

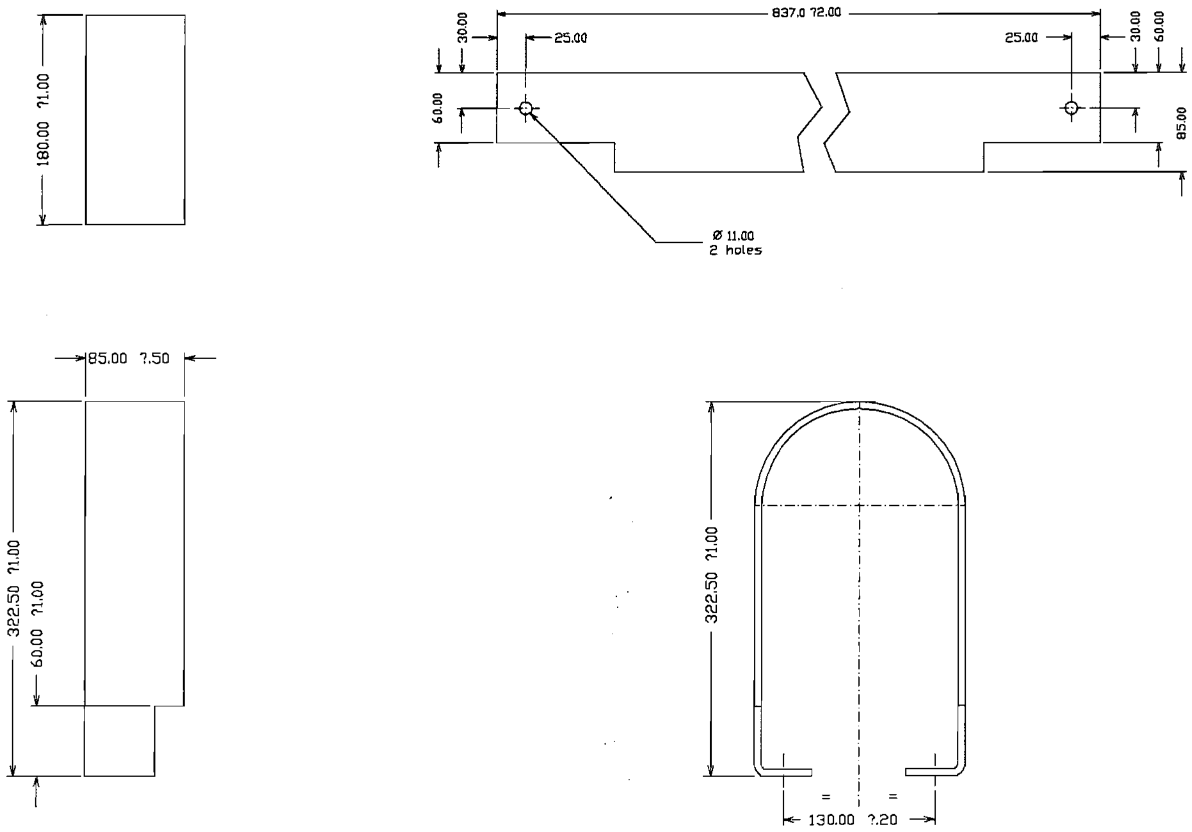
FLYWHEEL GUARD

SCALE: 1:2.5

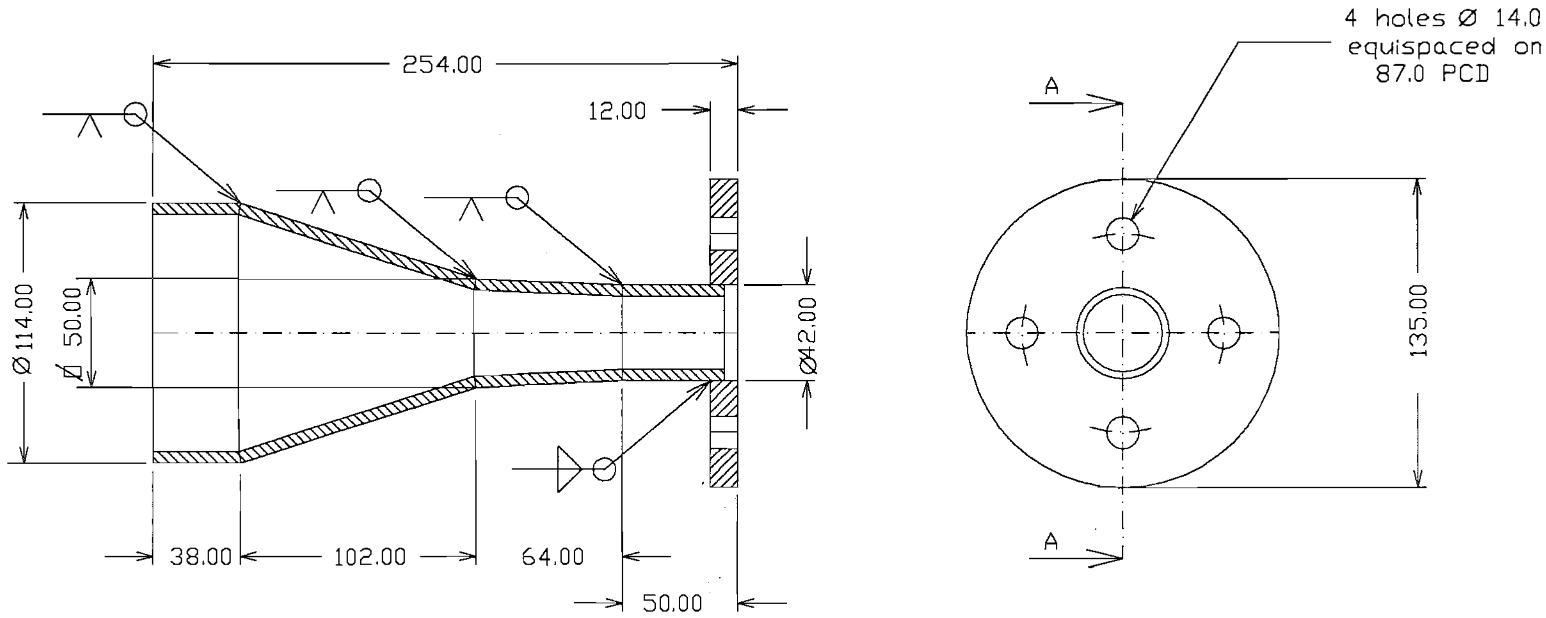
SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 8-8-90

DRG No: D400

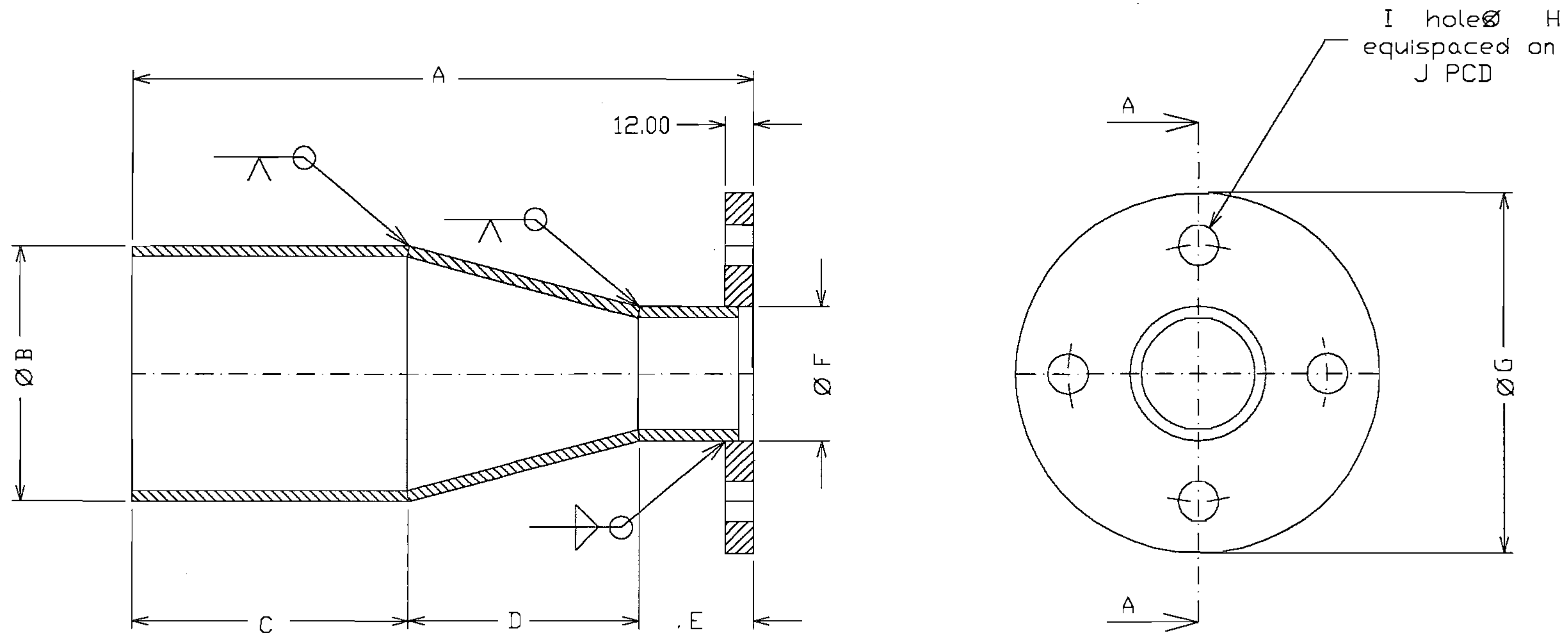


			ITEM	DESCRIPTION	QNTY
MATL: 6mm MILD STEEL PLATE		COUPLING GUARD	SCHOOL OF ENGINEERING MECHANICAL ENGINEERING DPT.		
THIRD ANGLE PROJECTION	A1		DRWN: S.HENG	DRG No: D410	
TOLERANCES: unless otherwise stated		SCALE: 1:2	CHKD:		
			DATE: 6-9-90		



SECTION A-A

		ITEM	DESCRIPTION	QNTY
MATL: Mild steel	TURBINE INLET ADAPTOR FOR 50x32-200		SCHOOL OF ENGINEERING MECHANICAL ENGINEERING DPT.	
THIRD ANGLE PROJECTION	A3		DRWN: S.HENG	DRG No: D500
TOLERANCES: ?.50 unless otherwise stated			CHKD:	
	SCALE: 1:2		DATE: 2-8-90	



SECTION A-A

MACHINES	A	B	C	D	E	F	G	H	I	J
1. 50x32-200			SEE DRAWING No. D500							
2. 65x40-200	254	114.0	122.0	102.0	50.0	48.0	145.0	14.0	4	98
3. 65x50-160	274	114.0	122.0	102.0	50.0	60.0	160.0	18.0	4	114
4. 80x65-160	305	165.0	115.0	140.0	50.0	76.0	180.0	18.0	4	127
5. 80x65-125	325	165.0	135.0	140.0	50.0	76.0	180.0	18.0	4	127

ITEM	DESCRIPTION	QNTY

MATL:
Mild steel

THIRD ANGLE PROJECTION | A3

TOLERANCES:
?.50
unless otherwise stated

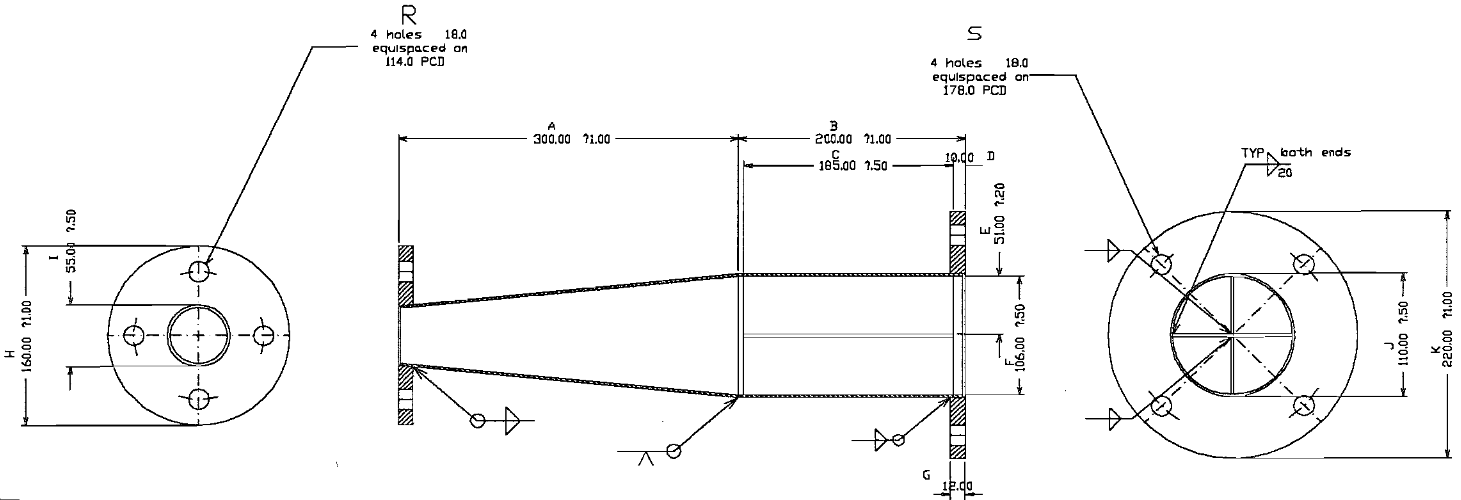
TURBINE INLET ADAPTOR

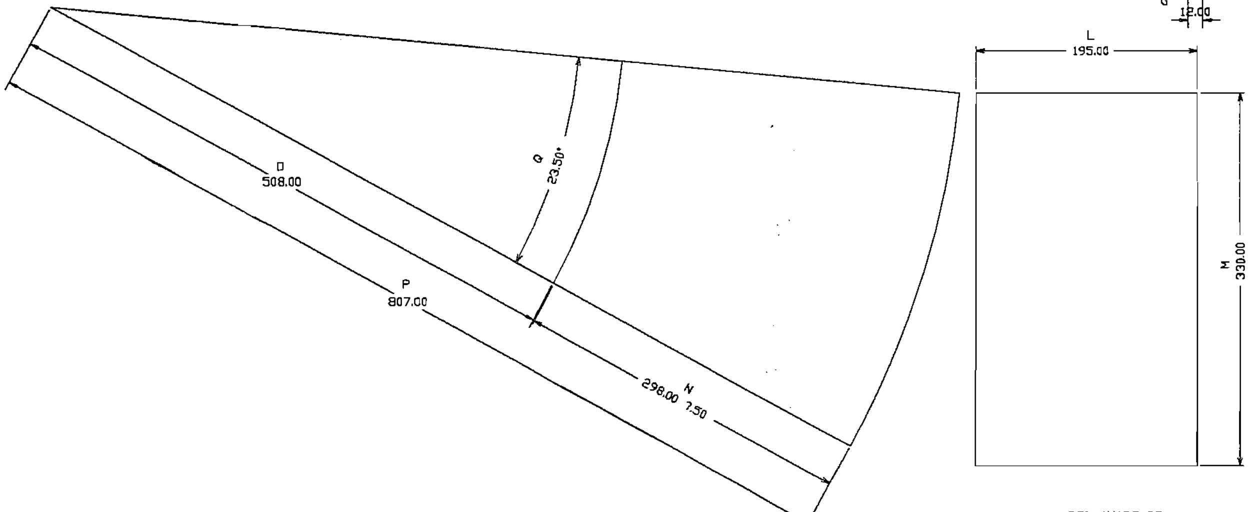
SCALE: 1:2

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 2-8-90

DRG No: D510





DEVELOPED SHAPE OF
50-100 DRAFT TUBE

DEVELOPED SHAPE OF
100mm FLOW STRAIGHTENER TUBE

DIMENSIONS

MACHINES	A	B	C	D	E	F	G	H	I
50x32-200	300	200	185	10	51	106	12	160	
65x40-200	300	200	185	10	51	106	12	160	
65x50-160	300	200	185	10	51	106	12	160	
80x65-160	600	300	285	10	77	156	12	200	
80x65-125	600	300	285	10	77	156	12	200	

MACHINES	I	J	K	L	M	N	O	P	Q
50x32-200	55	110	220	195	330	298	508	807	23.5
65x40-200	69	110	220	195	330	298			32.5
65x50-160	69	110	220	195	330	298			32.5
80x65-160	85	160	285	295	487	595			22
80x65-125	85	160	285	295	487	595			22

MACHINES		R
50x32-200	4 HOLES	18 EQUISPACED ON 114 PCD
65x40-200	"	" 127 PCD
65x50-160		" 127 PCD
80x65-160		" 146 PCD
80x65-125		" 146 PCD

MACHINES		S
50x32-200	4 HOLES	18 EQUISPACED ON 178 PCD
65x40-200		"
65x50-160		"
80x65-160	8 HOLES	18 EQUISPACED ON 146 PCD
80x65-125		"

ITEM	DESCRIPTION	QNTY

MATL: 2mm MILD STEEL SHEET

THIRD ANGLE PROJECTION | A1

TOLERANCES:
unless otherwise stated

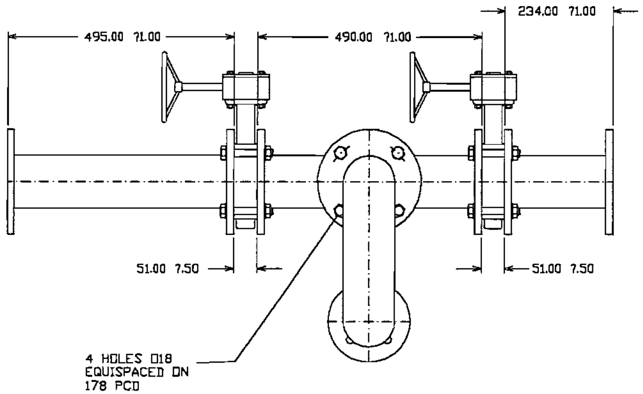
50mm, 65mm, 80mm
DRAFT TUBES

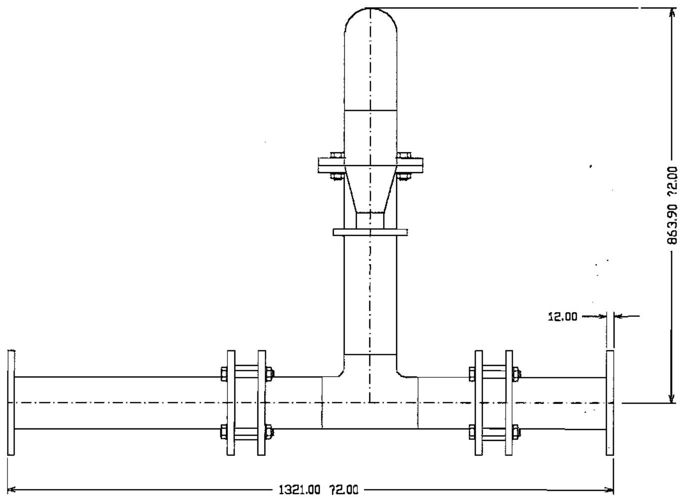
SCALE: 1 : 2

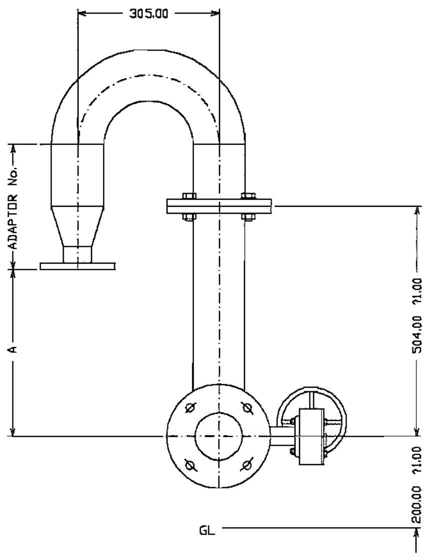
SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S HENG
CHKD:
DATE: 5-9-90

DRG No: D520







MACHINES	A	ADAPTOR No
1. 50x32-200	385.0	D500
2. 65x40-200	385.0	D510
3. 65x50-160	365.0	D510

NOTES

1) PIPE AND FITTINGS TO BS 4772

2) ALL JOINTS FLANGED, GASKET 3mm THICK
RUBBER, FULL FACED

3) ALL FLANGES TO BS 4504 PN 16 FLAT FACED

4) REFER TO DRG No D540 FOR MACHINES
4 80x65-160
5 80x65-125

ITEM	DESCRIPTION	QNTY

MATL: STEEL PIPES

THIRD ANGLE PROJECTION | A1

TOLERANCES:
unless otherwise stated

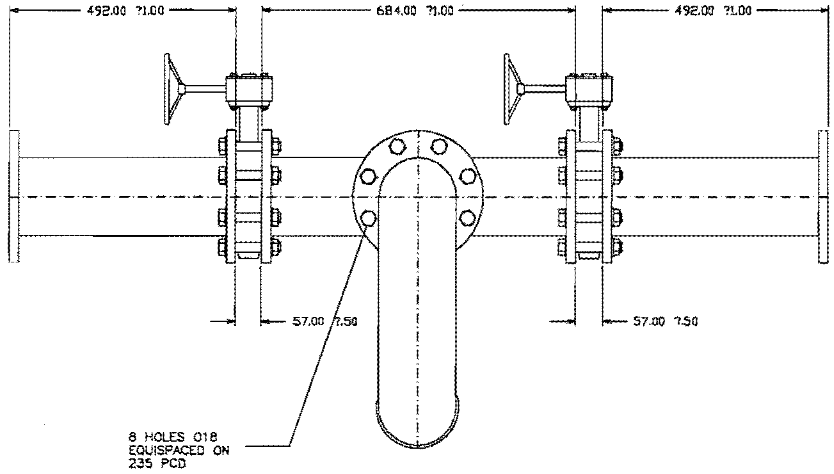
100mm PIPEWORK

SCALE: 1:5

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 8-8-90

DRG No: D530



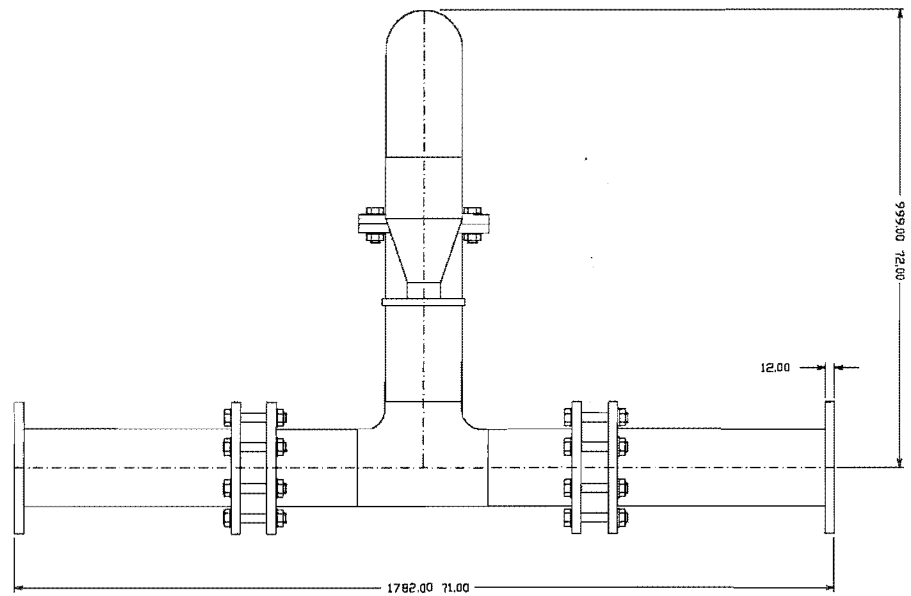
MACHINES	A	ADAPTOR No.
4. 80x65-160	385.0	D510
5. 80x65-125	365.0	D510

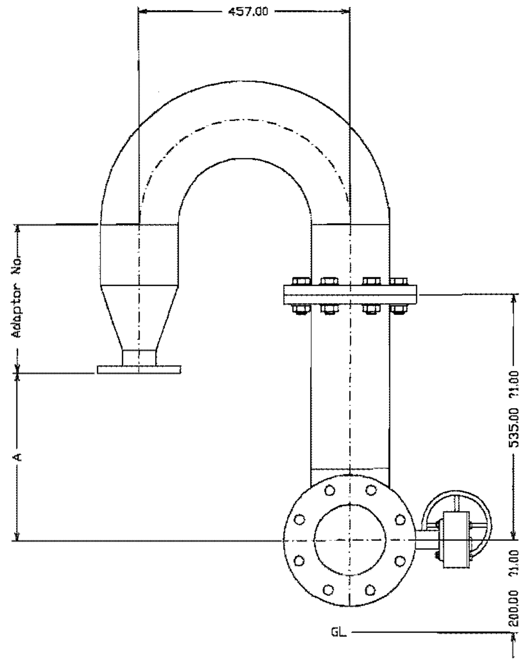
NOTES

1) PIPE AND FITTINGS TO BS 4772

2) ALL JOINTS FLANGED, GASKET 3mm THICK RUBBER FULL FACED.

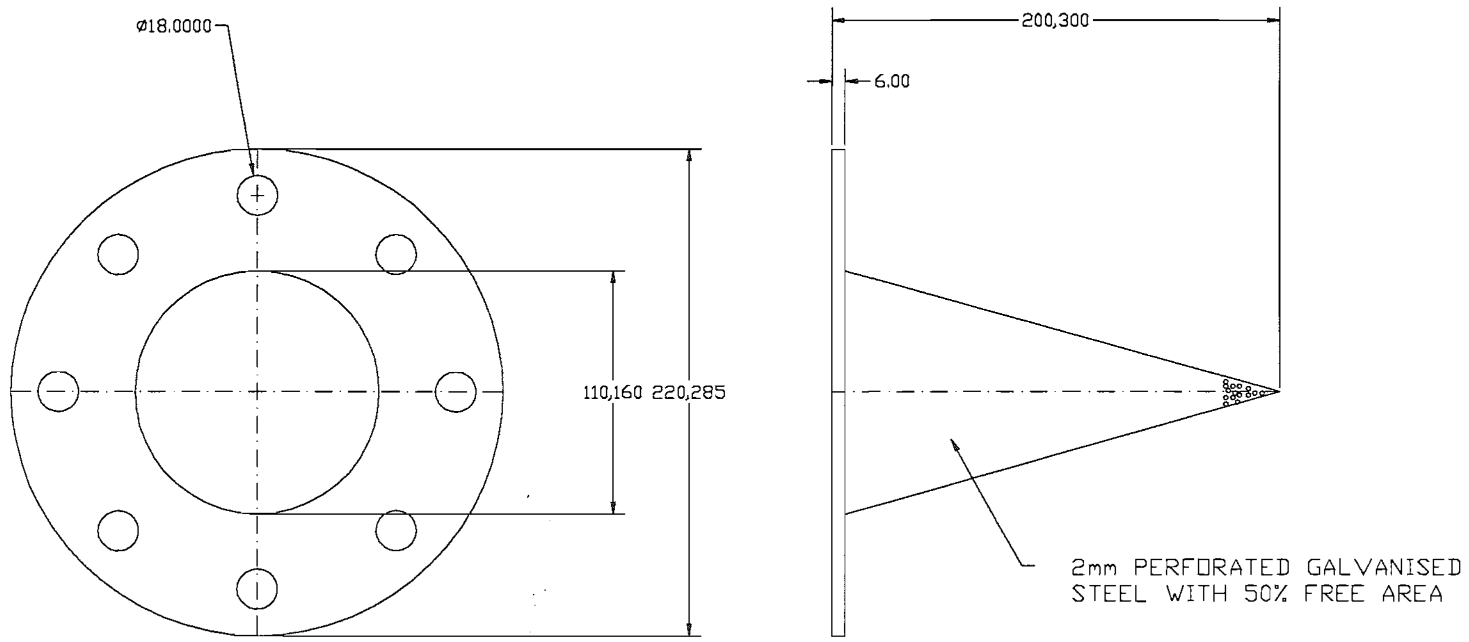
3) ALL FLANGES TO BS 4504 PN 16 FLAT FACED





ITEM	DESCRIPTION	QNTY

MATL: MILD STEEL	150mm PIPEWORK	SCHOOL OF ENGINEERING MECHANICAL ENGINEERING DPT.
THIRD ANGLE PROJECTION A1		DRWN: S.HENG CHKD: DATE: 3-9-90
TOLERANCES: unless otherwise stated	SCALE: 1:5	DRG No: D540



ITEM	DESCRIPTION	QNTY

MATL:
2mm MILD STEEL
GALVANISED

THIRD ANGLE PROJECTION | A3

TOLERANCES:
?.50
unless otherwise stated

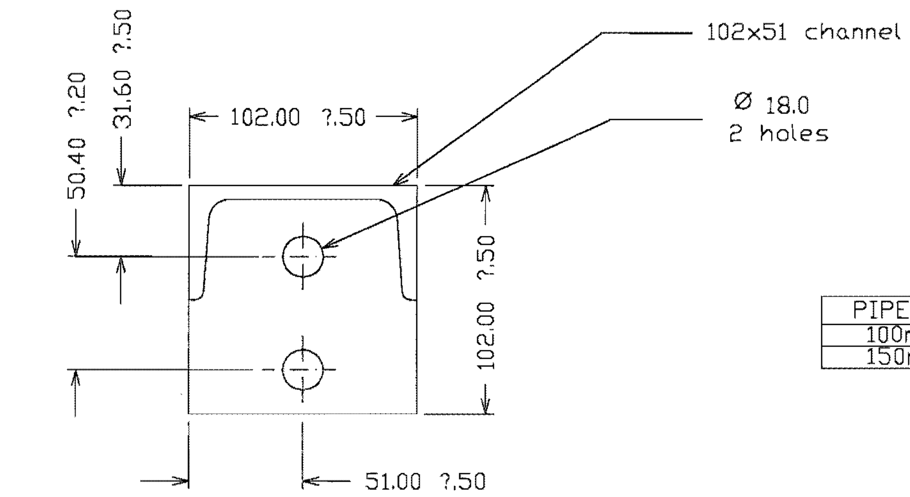
100mm AND 150mm
CONICAL STRAINER

SCALE: 1:2

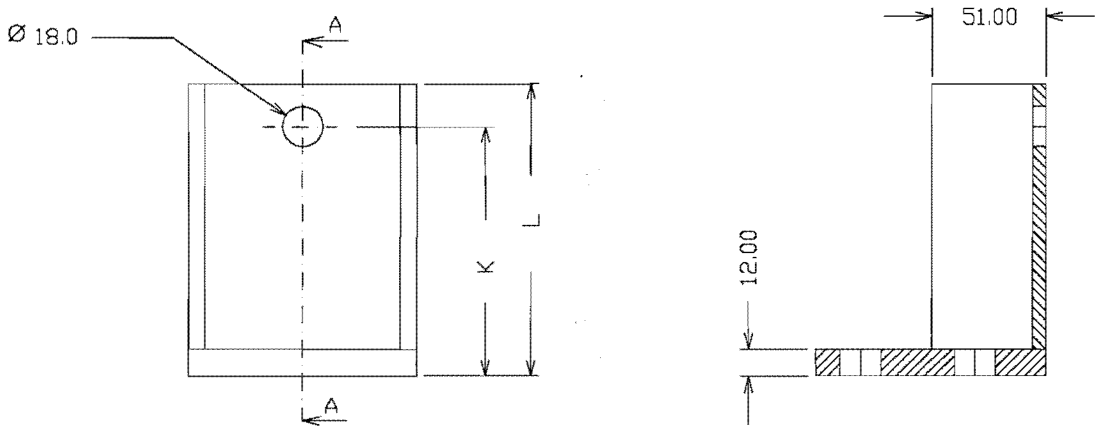
SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 2-8-90

DRG No:
D550



PIPE SIZE	K	L
100mm	111.0	130.0
150mm	71.5	91.5



ITEM	DESCRIPTION	QNTY

MATL:

MS channel and plate

THIRD ANGLE PROJECTION | A3

TOLERANCES:

unless otherwise stated

PIPE SUPPORT

SCALE: 1:2

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 8-8-90

DRG No:
D560

ITEM	DESCRIPTION	QNTY

MATL:

THIRD ANGLE PROJECTION A2

TOLERANCES:

unless otherwise stated

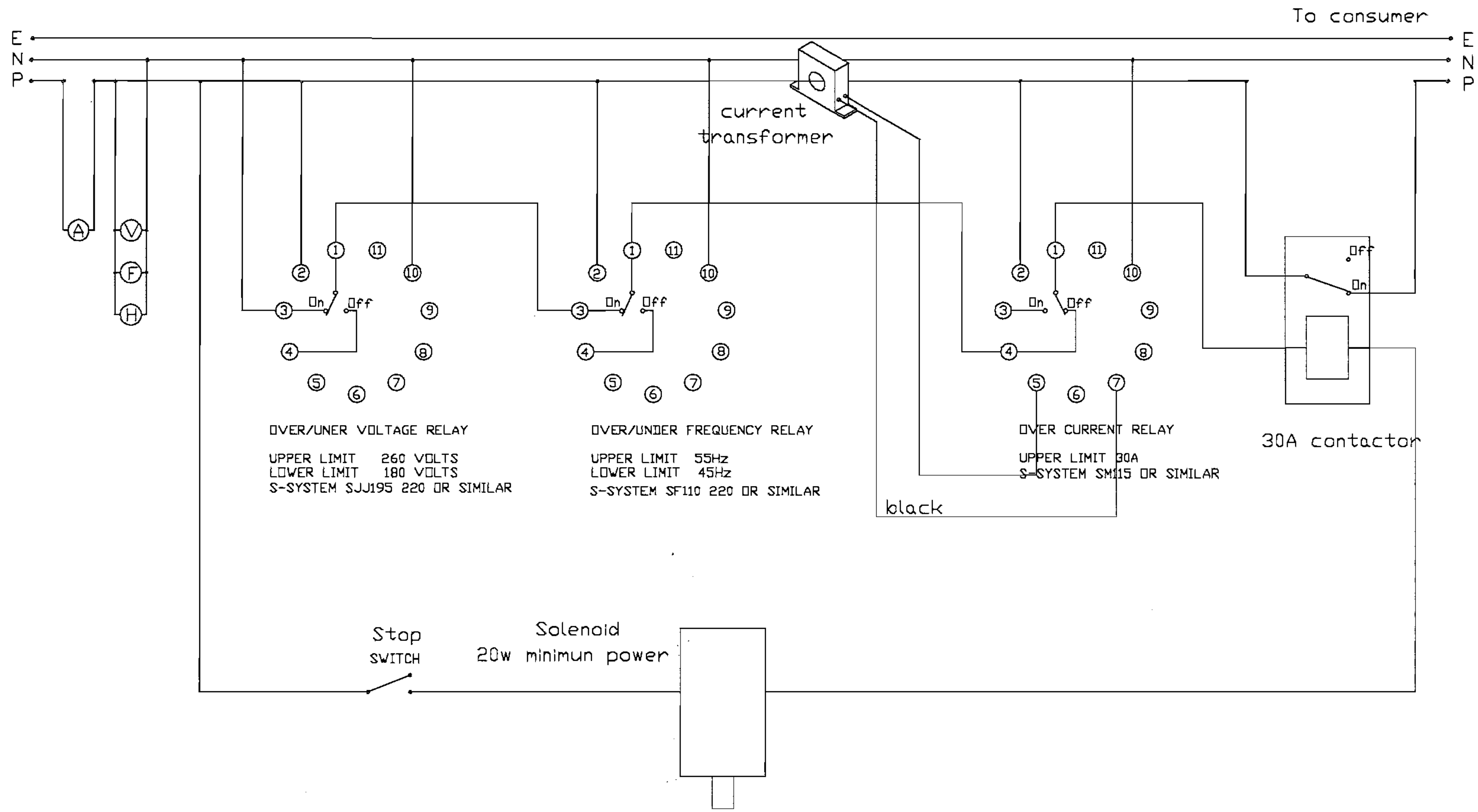
MALFUNCTION PROTECTION SYSTEM BOX

SCALE: 1:2

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 20-8-90

DRG No: D600



ITEM	DESCRIPTION	QNTY

MATL:

THIRD ANGLE PROJECTION | A3

TOLERANCES:

unless otherwise stated

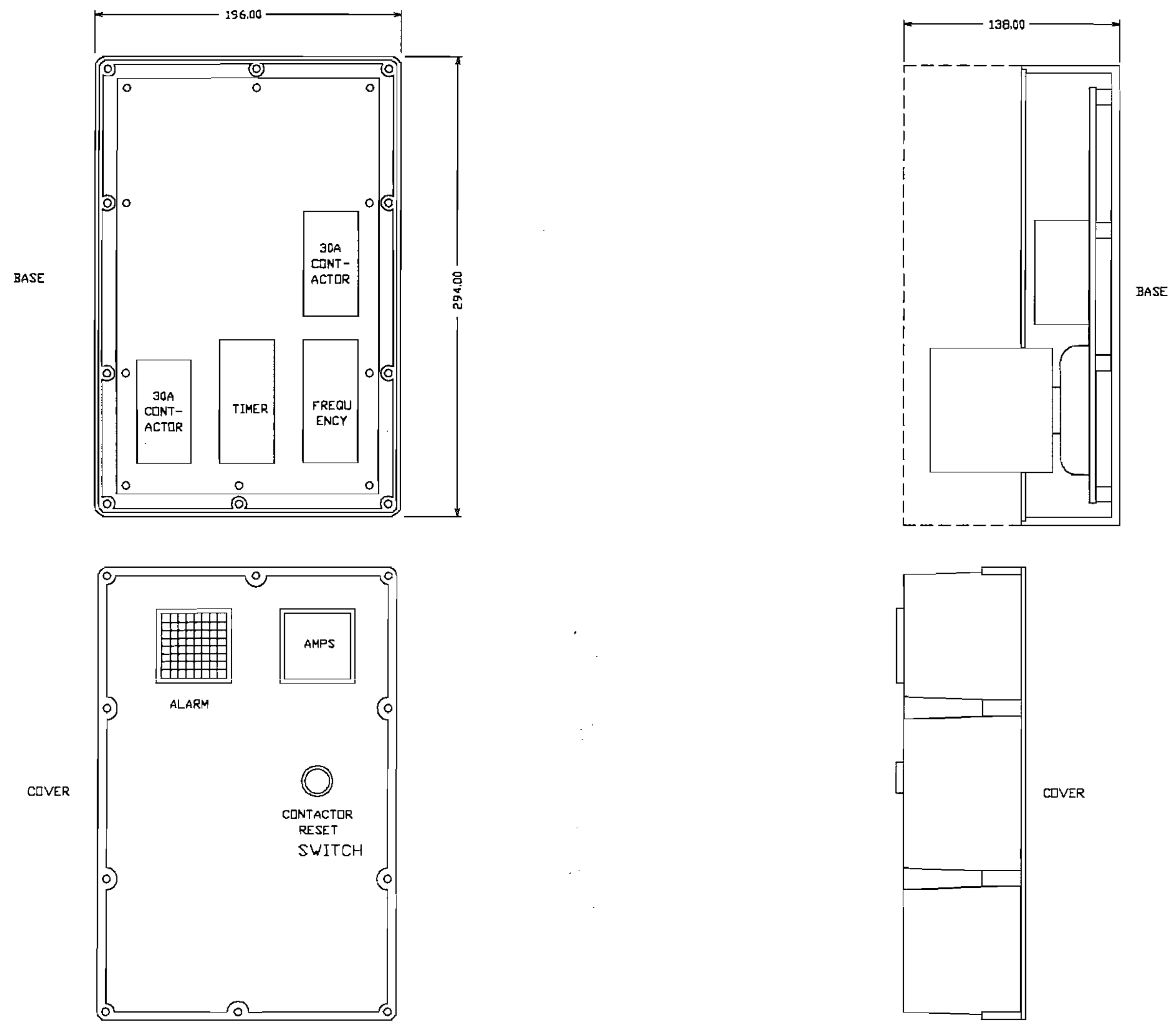
MALFUNCTION PROTECTION SYSTEM CIRCUIT

SCALE: NTS

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S . HENG
CHKD:
DATE: 5 JUNE 90

DRG No: D610



ITEM	DESCRIPTION	QNTY

MATL:

THIRD ANGLE PROJECTION | A2

TOLERANCES:

unless otherwise stated

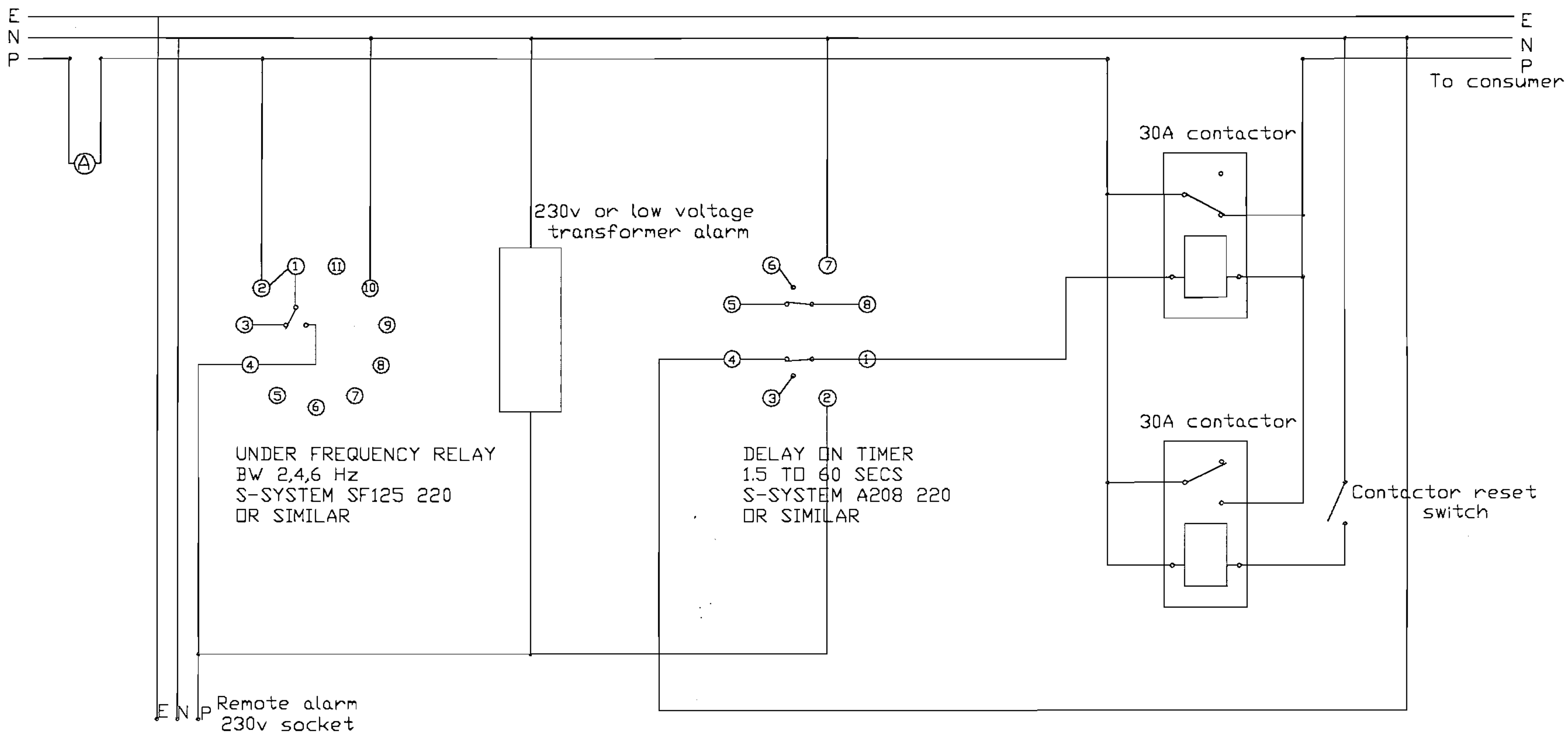
LOAD MANAGEMENT SYSTEM BOX

SCALE: 1:2

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:
DATE: 3-9-90

DRG No: D620



ITEM	DESCRIPTION	QNTY

MATL:

THIRD ANGLE PROJECTION | A3

TOLERANCES:

unless otherwise stated

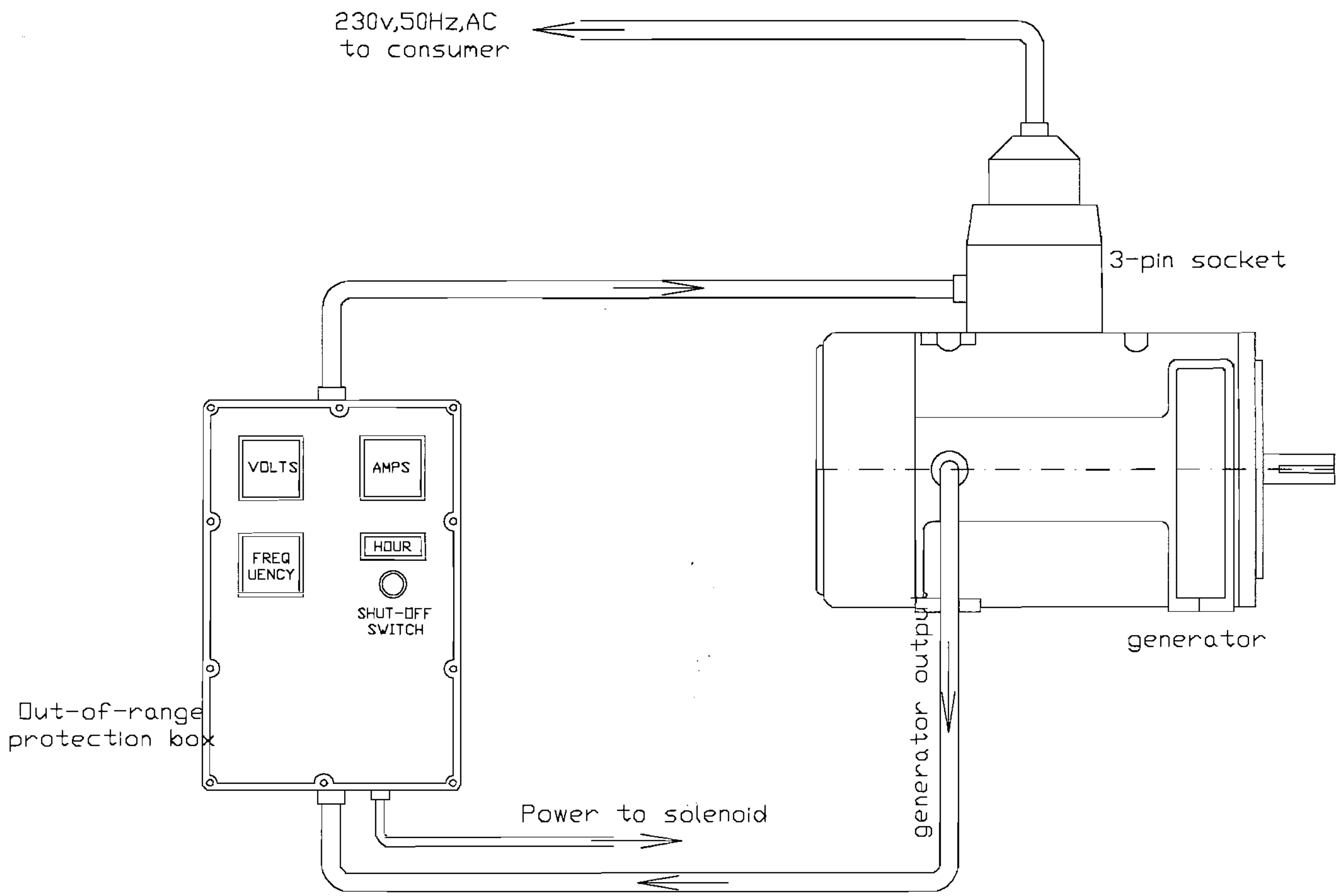
LOAD MANAGEMENT
SYSTEM CIRCUIT

SCALE: NTS

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S . HENG
CHKD:
DATE: 5 JUNE 90

DRG No:
D630



ITEM	DESCRIPTION	QNTY

MATL:

THIRD ANGLE PROJECTION — A3

TOLERANCES:

unless otherwise stated

GENERATOR WIRING ARRANGEMENT

SCALE: NTS

SCHOOL OF ENGINEERING
MECHANICAL ENGINEERING DPT.

DRWN: S.HENG
CHKD:

DATE: 20-8-90

DRG No: D640

APPENDIX 3

OPERATION AND MAINTENANCE MANUAL

OPERATION AND MAINTENANCE MANUAL

1 INTRODUCTION

The 5 kW Microhydro generating set is designed to harness potential energy in flowing streams to useful electrical output. At the designed condition, the machine will produce 4.8 kW of electrical output at 0.8 power factor (pf), 230 Volts, 50 Hz, AC.

Accompanying this document is the complete set of equipment which comprises the items outlined below. Supply and installation of the pipeline is the responsibility of the owner.

1 A baseplate on which the turbine and generator are mounted
2 Inlet and outlet pipework including valves and a draft tube
3 An electronic load governor and ballast resistor banks
4 Switchboards complete with instruments

This manual should be read and fully understood before the installation and finally running the plant. It should be retained for future reference as it contains information for regular maintenance.

2 SYSTEM DESCRIPTION AND OPERATION

There are five alternative arrangements of the equipment to cope with five different alternative site conditions. The machines operate on the "run-of-the-river" principle, diverting the necessary flow through a pipeline (penstock) to develop the head necessary to generate the electrical output. If there is less than the required flow and head, the equipment can still be run at less than full load. The system is autonomous, quiet, pollution free and requires very little maintenance.

A typical Microhydro installation comprises the following equipment:-

2.1 INTAKE

Water of suitable quality is extracted through the intake and fed down the penstock to a power house where the machinery is located. A small dam and a self-cleaning screen is recommended to extract the flow and separate out unwanted debris.

2.2 PENSTOCK

The penstock is the pipeline which delivers water from the intake to the turbine in the power house. Usually the penstock is sized such that the water velocity does not exceed 1.0 m/s to avoid excessive pressure surges and to minimise pipe friction.

2.3 PLANT ROOM EQUIPMENT

a) General mechanical layout

The machine in the power house converts the available power of the water to the corresponding electrical output. The unit comprises a centrifugal pump operating in the reverse mode as a water turbine (i.e. reversed direction of flow and reversed direction of rotation). Flow to the turbine is controlled by a geared butterfly valve located upstream of the turbine inlet. A second geared butterfly valve and a discharge pipe are provided for flushing the penstock. The turbine is directly coupled to a single phase, brushless, 2 pole, AC, synchronous generator with self induced excitation system and automatic voltage regulation. The generator produces a nominally constant voltage output within the 220-240 volt range, and frequency of 50 Hz (at 3000 RPM shaft rotation). Maximum power output from the generator is 6 kVA or 4.8 kW at 0.8 power factor.

b) Load governing

To maintain the constant frequency of 50 Hz, an electronic load governor (ELG) is used. The function of the ELG is to control the speed, and therefore frequency, of the small hydro plant by sensing the frequency and automatically controlling the total electrical demand presented to the AC generator. This is done by automatically diverting the unused electrical load to banks of dummy resistive load. The dummy loads may be resistor elements in air or water. The ELG may be located anywhere on the supply line, since frequency is the same throughout.

The switching of the ELG to control the frequency in response to the changes in the consumer load, is facilitated by the inertia of a flywheel mounted between the turbine and the generator. The flywheel's primary function is to provide extra inertia on the machine shaft to dampen out electrical transients experienced by the generator during the load switching. The flywheel is considered to be an integral part of the load governing process.

The flywheel's other function is to serve as a disc for the brake calliper. This feature is used to stall the machine in the event of a malfunction. The brake is held open when generating but released automatically by a malfunction protection system when a fault is detected. Stopping with the calliper brake is also the normal method of shutting down the plant.

c) Malfunction protection system

The malfunction protection system protects the generator and the appliances connected to the supply line from abnormal operating condition as the result of malfunction or breakdown. Relays sense the following electrical properties and release the brake and stall the machine if any one of those goes out of the set range.

i) Over/under voltage

Over and under voltage protection is necessary to prevent consumer appliances such as television, or other electronic equipment from operating at extreme voltages, and to protect motors from running at excessive current under conditions of low voltage. For the generator, extreme voltage can be caused only by failure of the voltage regulator. In this case, plant shutdown is appropriate. A voltage relay is used to initiate plant shut down if the voltage goes out of the set range of 180 to 260 volts.

ii) Over/under frequency

It is necessary to provide over and under frequency protection which can cause serious consequential damage to the generator and appliances connected to supply line. A frequency relay is used to initiate plant shut down if the frequency goes out of the set range of 45 Hz to 55 Hz. Over or under frequency, may occur as a result of one or a combination of the following:

1. Load rejection
2. Overload
3. Loss of excitation
4. Failure of AVR (Automatic voltage regulation)
5. Failure of ballasts load
6. Failure of ELG

iii) Over current

Over current protection is necessary to prevent excessive energy dissipation that would endanger plant or personnel as a result of equipment malfunction. The use of rewirable fuses is ineffective because the supply system is very unlikely to have the excess capacity to open them. A sensitive over current relay is therefore used with a setting of approximately 30 Amps.

NOTE *Do not re-start the machine until the faults have been investigated and remedied.*

d) Load management system

The load management system protects the consumer from nuisance tripping when trying to make full use of the power available. Should an overload occur, a warning alarm comes on for 20 seconds. If overload continues, (perhaps due to absence of the consumer) then the supply to the consumer will be disconnected. All the power is then automatically diverted to the dummy loads. To resume supply, the consumer needs to reduce the load in response to the alarm and thus remove the overload condition. The alarm is automatically cancelled and the timer reset when the consumer is disconnected.

Overload can occur in various ways as for example the when consumer can unknowingly switch on excess load, or a thermostat can switch on excessive load automatically.

NOTE *Ensure that the load is reduced before reconnecting the consumer to the supply. Reconnection is done by pressing the contactor reset button located on the overload warning box.*

3 INSTALLATION

3.1 FOUNDATION

The foundation must be of sufficient size and rigidity so as to prevent movement when the machine is anchored down, and to absorb vibration when the machine is running. A well laid concrete slab is the most effective way of ensuring a sound foundation.

The foundation should be no less than 150 mm thick with steel reinforcement to prevent cracking. The foundation bolts should preferably be installed as shown in Fig A3.1. This is to prevent movement when the baseplate is bolted down. The use of a template is recommended for setting out the foundation bolts.

NOTE *Take care not to tighten down the foundation bolts excessively or the baseplate may distort and upset the alignment of the machine. Grouting of the baseplate to the foundation is optional.*

3.2 PIPEWORK

The inlet pipework is supported on the foundation independently of the machine. This will allow for the stresses and any unbalanced forces in the pipework due to pressure surges, to be taken up by the independent pipe support. The turbine and pipe flanges must be parallel and mate together without the use of force. This is to ensure that abnormal strain is not transmitted to the turbine casing. All pipework has been pre-assembled in the factory to ensure correct fitting of all parts.

NOTE *Do not force alignment of flanges or bolt holes as this could damage the turbine and throw-out turbine/generator alignment.*

To avoid unnecessary strain on the outlet side of the turbine, light pipe only needs to be used to connect to the standard draft tube. The draft tube and the associated outlet pipework must be supported. The flushing outlet pipework must also be supported.

3.3 PROTECTION EQUIPMENT MOUNTING

There are three major pieces of control and protection equipment associated with the microhydro plant:-

1 Electronic load governor
2 Malfunction Protection system
3 Load management system

The Electronic load Governor may be located at any convenient location on the supply line. Normally, it is located near where the dump load can be utilized. (i.e for water or space heating of a green house or animal sheds). Ensure that the dump loads are properly enclosed and adequately ventilated to avoid fire hazard due to radiant heat. It is recommended that a furnace type enclosure is used. Refer to operating manual of the Load Governor for connection of dump loads.

The malfunction protection equipment is located on the machine. This is to ensure that protective response signal can easily be accessed.

The load management system may be located at any convenient location on the supply line. Normally, it is located at the consumer end. This is to enable the power used to be monitored and allow easy access for the audio and/or visual warning alarm.

4 PRELIMINARY CHECK

Before starting the newly installed unit, carry out the following procedures:-

1 Check the brake operation by manually pulling the lever and releasing it.

2 Pull the brake open and turn the machine shaft by hand to ensure that it is running freely. If this is not the case, the cause of the problem will need to be investigated.

3 Check the electrical connections to the generator and to the release solenoid. Make sure the 3-pin plug is fully home in the socket and the locking ring screwed up firmly.

NOTE *The turbine is fitted with a mechanical seal and should not be rotated unnecessarily when dry. Care must be taken not to damage the seal faces.(Refer to KL-ISO pump manual for mechanical seal replacement procedure).*

5 INITIAL START-UP

5.1 PIPELINE FLUSHING PROCEDURE

The following procedures are to be carried out and items checked before starting.

1 Flush the penstock after installation but before it is connected to the valves or the turbine.

2 Fill and flush the penstock after all connections are made. Use the pressure gauge as a guide and make sure the pressure surge does not exceed the maximum limit. Close the valve to allow the pipeline to fill up after flushing.

NOTE

- *Operate the valve slowly and allow approximately 1 minute for the whole operation. Be extra cautious toward the last 10% of the closing operation and the first 10% of the opening operation.*

- *Do not operate the valve without the pressure gauge.*

3 Check the brake setting by gradually opening the valve with the brake on. The brake must be able to just hold the machine stalled at full head. Adjust the brake spring tensioning screw if necessary.

NOTE *The brake will heat up with repeated use during the initial set up, this is normal. The bearings will initially heat up but will eventually reduce to normal warm operation. The generator normally runs hot.*

6 START-UP

6.1 THE STARTING PROCESS

At starting, the machine accelerates rapidly from rest and reaches full voltage as it approaches synchronous speed. The frequency control equipment rapidly energizes and holds the machine speed at synchronous speed.

6.2 THE STARTING PROCEDURE

1 Initially, the machine is held stationary by the brake. The inlet valve is then opened slowly. The brake torque needs to be sufficient to hold the machine stalled in this extreme condition.

2 Manually pull the brake open and the machine will accelerate to synchronous speed and where it will be held there by the load governor. The malfunction protection equipment will come on to hold the brake open. The brake lever can then be released.

Check that the machine is running smoothly and the voltage, current and frequency are nominally correct. Use the emergency stop switch to stall the machine if necessary.

7 STOPPING

To stop the machine, press the emergency stop switch. This disconnects the power to the solenoid and releases the brake, bringing the machine to rest. Water will continue to flow through the turbine. The valve can either be closed carefully or left open as water flowing through the stationary pump turbine causes no harm.

During the braking operation, the sudden change in hydraulic resistance of the centrifugal pump as it is stalled is small and not sufficient to induce any serious water hammer. It should be noted however that, the microhydro generating set is designed to run continuously, but can be stopped for repairs and maintenance or may stop due to the electrical faults in the house or the control circuits.

8 MAINTENANCE

The following should be checked periodically for wear or damage and replaced if necessary.

8.1 INTAKE

The state of the intake can be judged by the pressure gauge at the turbine inlet. Partial blockage of the intake or lack of water will cause the pressure reading to drop below normal operating value. Check and clean the intake regularly. The period of maintenance will be determined by experience as this is dependent on the site.

8.2 PENSTOCK

The penstock should be flushed regularly by having both valves fully opened. Check and clean the conical strainer periodically. This is done by slowly closing the inlet valve, and opening the flush valve to drain off excess water. Remove the U-bend pipe by carefully unbolting it from the turbine. Clean the strainer then refit the pipework. DO NOT force alignment of flanges as this could throw out the turbine-generator alignment.

8.3 TURBINE

Check the state of the turbine periodically. (i.e. the bearings should be quiet and not overheating). Check for drips in the turbine seals, (the mechanical seals should not drip-if they do a replacement is need). Any water leaking out should be thrown off by the water thrower. Replace bearings and mechanical seals annually.

8.4 GENERATOR

In general, periodically check the generator for noise and overheating, and replace the bearings annually. DO NOT have the light continuously on in the generator shed as the insects attracted by the light will be drawn into the generator by its cooling fan.

8.5 DUMMY LOADS

Check the state of the dummy loads periodically. For radiant elements, check for accumulation of dust which may restrict the air flow over them. For elements in water, occasionally inspect for build up of lime.

8.6 BRAKE

The operating mechanism should be tested weekly using the emergency stop switch. Check the pin joints, lubricate if necessary. Check brake pad surfaces annually. Refer to the KL-ISO pump manual for the annual overhaul of the turbine and the generator before replacing the bearings and mechanical seals.

9 SPECIFICATIONS

9.1 ELECTRICAL CAPACITY

Voltage	Single phase, 230 volts, AC
Frequency	50 Hz (Nom), Droop 49.3 - 50.8 Hz
Power	5 kW (Nom), 4.8 kW (Actual) at 0.8 pf

9.2 TURBINE

Performance of the KL-ISO centrifugal pumps operating as turbines is outlined in Table A3.1. All the machines operate at 3000 RPM producing shaft power output of 6 kW.

MACHINE	NET HEAD(m)	FLOW (l/s)	PIPE (mm)
50x32-200	155	9.0	100
65x40-200 *	81	14.0	100
65x50-160	64	14.5	100
80x65-160 *	42	21.0	150
80x65-125	27	28.0	150

Table A3.1 Turbine performance parameters

* The impellers have been trimmed to produce the required shaft power at peak efficiency - refer to performance graph for detail.

9.3 GENERATOR

The generator is a 2 pole, synchronous, brushless generator, with self excitation and automatic voltage regulation.

Type	Markon B21D
Voltage	Single phase 230 volts, 50 Hz, AC
Power	6 kVA or 4.8 kW at 0.8 pf

9.4 FREQUENCY (SPEED) CONTROL

An electronic load governor rated 20 kW, Frequency droop 49.3 - 50.8 Hz, Single phase 230 volts, 50 Hz, AC.

9.5 MALFUNCTION PROTECTION SYSTEM

ITEMS		SETTING
VOLTAGE (UL Upper level) (LL Lower level)	UL	260 V
	LL	180 V
FREQUENCY	UL	55 Hz
	LL	45 Hz
CURRENT	UL	30 A
	LL	

Table A3.2 Malfunction protection system setting range

9.6 LOAD MANAGEMENT SYSTEM

ITEMS		SETTING
FREQUENCY	LL	48 Hz

Table A3.3 Load Management System

10 LIST OF MANUFACTURERS AND SUPPLIERS

This following is a list of major manufacturers and suppliers of parts for the Microhydro generating set.

1 B R HOMERSHAM LTD
PO Box 280
155 Roydvale Ave
Christchurch Tel (03) 358-8309, Fax (03) 358-2516

General Engineering firm, specialising in irrigation. Manufacturer of both diesel and electric fire pump. Supplier of KL-ISO centrifugal pumps, and the Microhydro generating set.

2 POWER EQUIPMENT LTD
3 Emerdale lane
Christchurch Tel (03) 379-3135

Specialist in portable power generation, and supplier of Markon Brushless B21D synchronous generator.

3 DIESEL AND POWER SYSTEMS NZ LTD
P O Box 359
45 Reid Rd
Dunedin Tel (03) 555-946, 552-999, Fax (03) 555-401

Consultants, Generator specialist, Diesel, Hydro and Marine Engineers. Manufacturer of the 20 kW Electronic load Governor.

4 PAYKEL ENGINEERING
P O Box 382
96-98 Tuam St,
Christchurch Tel (03) 379-0120

Engineers supplier, Supplier of the Twiflex MSF mechanical calliper.

5 FENNER-BURNS LTD
61 Coleridge St,
Christchurch Tel (03) 365-0884, Fax (03) 365-0906

Engineers supplier, Supplier of Fenaflex coupling F40, SM16 spacer coupling.

6 J F HARGRAVES LTD
P O Box 1037
173 Wordsworth St,
Christchurch Tel (03) 379-6510

Specialist in flow control equipment. Supplier of geared butterfly valves.

7 STEEL AND TUBE NZ LTD
Blenheim Rd
Christchurch Tel (03) 348-2019

Supplier of steel pipes and pipe fittings.

8 FLETCHER STEEL
P O Box 8372
Lunns Rd
Christchurch Tel (03) 348-8479

Stockist and supplier of mild steel, Reinforcing steel etc.

9 NZ SOLENOID CO LTD
P O Box 2405
196 St Asaph St
Christchurch Tel (03) 366-6365, Fax (03) 379-6590

Rotary switches, motor control equipments. Supplier of power control relays, protection system etc.

10 REDPATH R LTD
312 St Asaph St,
Christchurch Tel (03) 379-0446

General Electrical supplier. Supplier of power relays, protection system etc.

11 AVON ELECTRIC LTD
185 Dyers Rd
Christchurch Tel(03) 384-4139

Supplier of the Dummy load banks

Fig A3-1

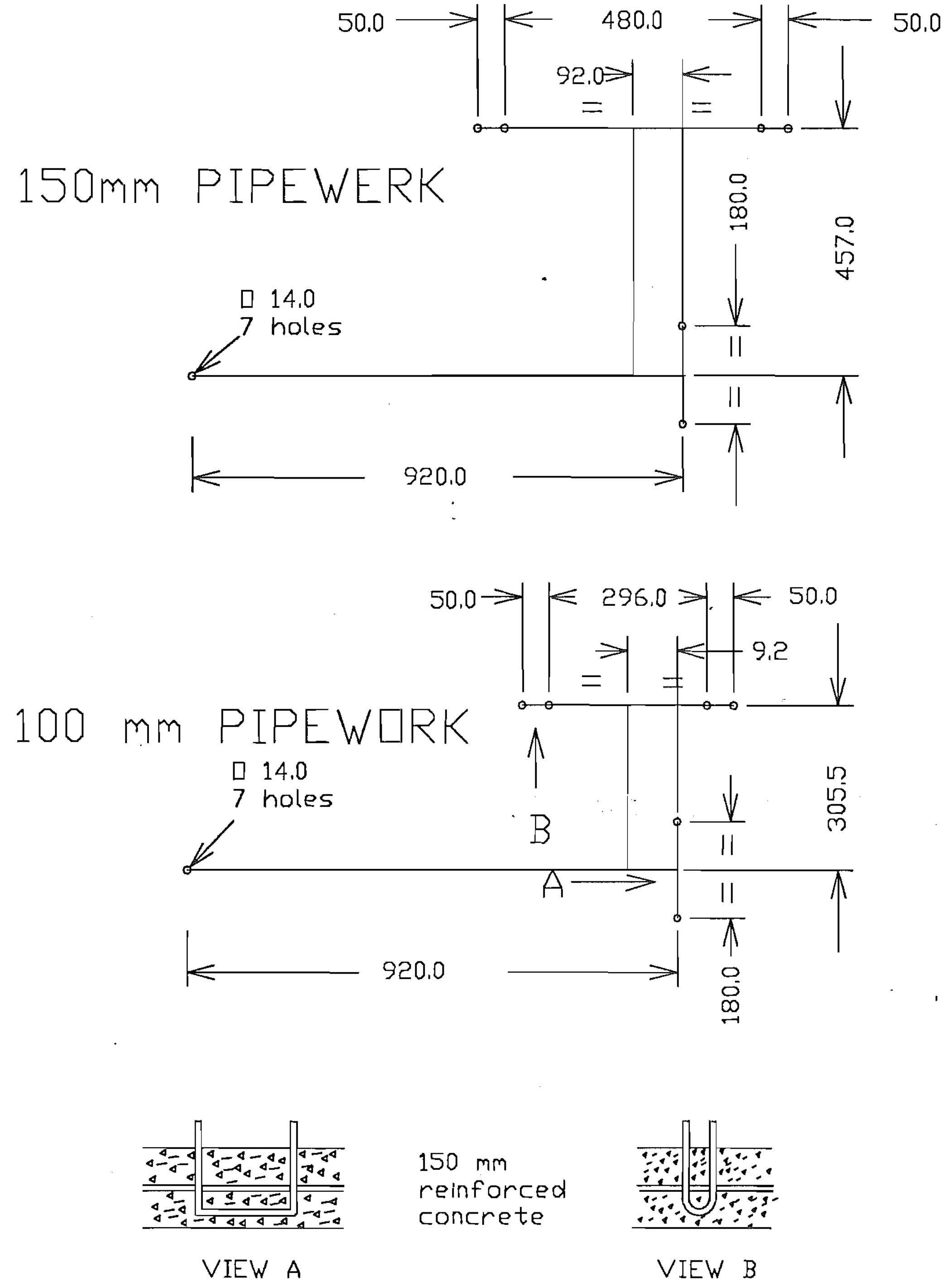


Fig A3-1 Foundation bolts

APPENDIX 4

DETAILED CALCULATIONS

DETAILED CALCULATIONS

1 THE TUBINE SHAFT

1.1 SHAFT STRESS CALCULATIONS

For turbining, shaft stress calculations are necessary due to the unusual operating conditions as outlined below.

1. Reversed direction of shaft rotation
2. Reversed direction of water flow
3. Increased head and capacity to more than double the pumping case
4. Introduction of additional stresses on the shaft due to deadweight of flywheel, and impact stresses due to sudden braking

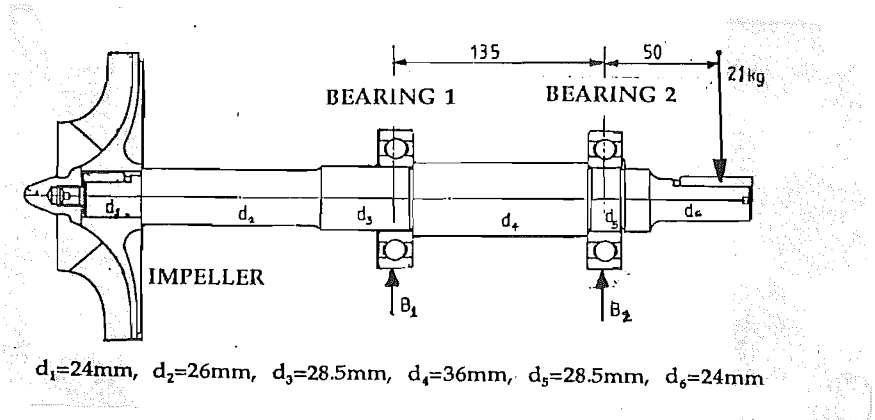


Fig A4.1 The turbine shaft

1.2 STRESS CALCULATIONS BASED ON FATIGUE EVALUATION

The shaft material is Stainless steel 431-S29

Ultimate tensile stress	F_u	900 MPa
Yield stress	F_y	680 MPa
Endurance stress	$F_{e'}$	635 MPa

The shaft will be subjected to the following stresses:-

1. Tension
2. Compression
3. Bending
4. Torsion

No axial load will be assumed for this application, because the impeller is arranged to minimise axial thrust.

From Labanoff [23], the equation for acceptable loading,

$$\left(\frac{\tau_a}{.5nF_e} + \frac{\tau_m}{.5nF_y}\right)^2 + \left(\frac{\sigma_a}{nF_e} + \frac{\sigma_m}{nF_y}\right)^2 \leq 1 \qquad \text{(EA4-1)}$$

The primary loads of the turbine shaft are generally torsion and bending loads. A safety factor is usually applied to F_e and F_y to account for unanticipated loads. (EA4-1) is applied at the location(s) where stresses are the highest.

For loading conditions more than 1000 cycles, fatigue analysis is necessary. Therefore, it is necessary to calculate fatigue stress analysis due to weight of flywheel and coupling hanging on the stub shaft of the turbine.

a) Fatigue analysis

Properties of the flywheel

Diameter	= 470 mm
Mass	= 16 kg
Inertia	= 0.45 kg/m2

Mass of coupling and spacer etc
= 5 kg

Therefore, total mass of fittings overhanging the stub shaft is
= 21 kg

The centrifugal pump impeller is designed to have a well balanced force hydraulically. Vanes located behind the impeller balance out the hydro static pressure. It is assumed that the above still applies for turbining operation. Therefore, the only unbalanced force cause by the water is the torque acting on the shaft.

Forces and bending moments on the shaft due to weight of the flywheel and coupling are as follows:-

Assumptions:

- No hydraulic load as outlined above
- Neglect weight of the shaft
- Neglect weight of the impeller

Referring to Fig A4.1, the forces and bending moments are as follow, with the greatest shear force occur at bearing B2.

B_1	= -76.3 N
B_2	= 282.3 N
BM	= -10.3 Nm

b) Torsional and bending stresses

i) Torque

Assuming shaft power rating of 6 kW, the torque on the shaft is,

$$T = \frac{P}{\omega} = \frac{6000}{314.2} = 20.7 \text{ Nm}$$

ii) Alternating bending stresses

$$\sigma_a = \frac{(BM)r}{I}$$

$$I = \frac{\pi D^4}{64}$$

Where,

m = 10.3 Nm
D = 24 mm at coupling, 30 mm at bearing

$$I = \frac{\pi(.03)^4}{64} = 3.98 \times 10^{-8} \quad m^4$$

$$\therefore \quad \sigma = 3.88 \text{ MPa}$$

The endurance limit for 431-S29 steel must be adjusted to account for service conditions.

$$F_e = K_a K_c K_e F_{e'}$$

$$F_e' = 635 \text{ MPa}$$

Where K_a, K_c, K_e, are factors with the following definitions:-

For 99% reliability $K_c = 0.814$
For ground surface finish $K_a = 0.9$

For stress concentration of shape neck, where the bearings sit

$$Ke = \frac{1}{K_t} = \frac{1}{1 + q(K_t - 1)}$$

In dynamic mode,

q = 0.8
K_t = 2

$\therefore \quad K_e = 0.556$

$\therefore \quad F_e = (0.9)(0.814)(0.556) \times 635$

$= 259$ MPa

iii) Torsional stress

$$\tau = \frac{Tr}{J}$$

$$J = \frac{\pi D^4}{32} \qquad \text{(at D = 30 mm)}$$

$$= 7.95 \times 10^{-8}$$

$$\tau = 3.9 \text{ MPa}$$

Since there is no axial component, the equation for acceptable loading (EA4-1) becomes :-

$$\left(\frac{\tau_m}{.5nF_y}\right)^2 + \left(\frac{\sigma_a}{nF_e}\right)^2 \leq 1 \qquad \text{(EA4-2)}$$

Applying a safety factor of n = 1.5 to F_y and F_e,

$$\left(\frac{3.9}{.5\times1.5\times680}\right)^2 + \left(\frac{3.9}{1.5\times259}\right)^2 \leq 1 \qquad \text{(EA4-3)}$$

$$0.0008 \leq 1$$

Based on the above results, it can be concluded that the flywheel can safely be overhung without risk of fatigue failure.

1.3 FORCES AND TORQUE DURING BRAKING

The braking operation produces a reaction force and a reaction torque on the turbine shaft in the direction as shown in Fig A4.2. Owing to shock associated with sudden braking, the reaction force and torque may be a number of times greater than the applied force. However, it is considered that braking operations occur less than 1000 times for the life time of the machine. Therefore, it is necessary to calculate for static strength only.

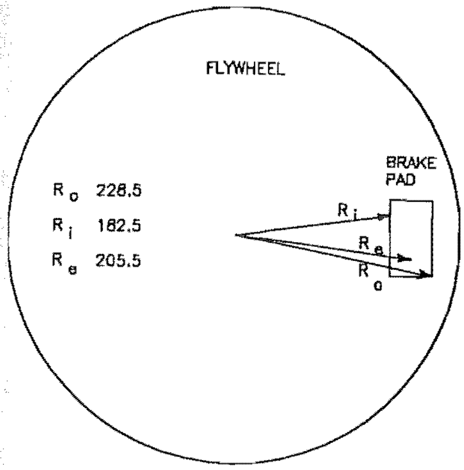


Fig A4.2 Flywheel/Brake

In order to bring the machine's shaft to rest during normal operation, the brake must overcome the inertia of the flywheel and all the rotating components as well as the torque from the turbine.

Torque to stop shaft (due to inertia of the flywheel) i.e. Torque to bring rotating machine from speed ω_0 to lower ω_f (=0) in a slow down time of t_s is given by:-

$$T = \frac{I(\omega_o - \omega_f)}{t_s}$$

Where,

$\frac{I(\omega_o - \omega_f)}{t_s}$ represent the change in kinetic energy,

Torque required to bring the shaft to rest due to flywheel inertia is,

$$T = \frac{0.45\,(314.2)}{5}$$
$$= 28\ \text{Nm}$$

Torque to stop shaft (due to turbine power)

ω = 314.2 rad/sec (or 3000 rpm)
P = 6.0 kW

$$T = \frac{6000}{314.2}$$
$$= 19\ \text{Nm}$$

Total torque require to bring shaft to stop is,

$$T = 28 + 19$$
$$= 47 \text{ Nm}$$

At stall flywheel inertia torque is zero, and the operating torque, is twice that of the torque at the operating speed of 3000 RPM. i.e. 38 Nm.

From Fig A4.2, the following can be defined (assuming uniform wear over pads):-

$$R_e = \frac{(R_i + R_o)}{2}$$

Torque capacity per pad is given by:-

$$T = \mu F' R_e$$

Therefore, the torque capacity for 2 pads is shown below. This is the torque required to bring the machine to a stall from normal operation:-

$$T = 2\mu F' R_e$$

Force for both pads required to stop shaft is,

$$F' = \frac{T}{2\mu R_e}$$

$$F' = \frac{47}{(2)(.37)(.21)}$$
$$= 302 \text{ N at each pads}$$

a) Reaction force on the shaft

From Shigley, the average and maximum contacting pressure

$$P_{max} = \frac{F'}{(\theta)(R_i)(R_o - R_i)}$$

$$\frac{P_{avg}}{P_{max}} = \frac{2(R_e/R_o)}{1 + R_i/R_e}$$

In the limit the average and maximum pressure becomes equal as R_i ---> R_o

$$R_i/R_o = 0.6$$

$$P_{avg} = 0.75\, P_{max}$$

$$R_i/R_o = 0.8$$

$$P_{avg} = 0.89\, P_{max}$$

The uniform wear assumption is conservative for a given actuating force. It implies a smaller torque capacity.

Maximum contact pressure (F = 2 pads)

$$P_{max} = \frac{302}{(25/57.3)(.1825)(.228 - .1825)}$$
$$= 83.4 \text{ kPa for both pads}$$

$$P_{avg} = (0.89)(83.4)$$
$$= 74.3 \text{ kPa for 2 pads}$$

Reaction force on shaft when both brake pads applied

$$F = 2\mu F'$$
$$= (0.37)(302)$$
$$= 223 \text{ N}$$

Reaction torque when brake applied

$$T = FR_e$$
$$= (223)(.2055)$$
$$= 46 \text{ Nm}$$

The reaction toque is therefore the torque capacity for 2 pads.

Apply the following shock factors (From Hosking and Harris)

K_m = 3 For bending
K_t = 3 For torsion

The reaction force and torque now become,

F = 669 N
T = 138 Nm

By force equilibrium

B_1 = 248 N
B_2 = -917 N
BM = 33.5 Nm

b) Von Mises stress analysis

The Von Mises stress equation for combined loading and torsion.

$$\sigma' = (\sigma^2 + 3\tau^3)^{\frac{1}{2}} \tag{EA4.4}$$

Where,

$$\tau = \frac{16T}{\pi d^3}$$

$$\sigma = \frac{Mc}{I} = \frac{32M}{\pi D^3}$$

$$I = \frac{\pi D^4}{64}$$

$$J = \frac{\pi D^4}{32}$$

Torsional stress,

$$\tau = \frac{(16)(138)}{\pi(0.03)^3}$$

$$= 26 \text{ MPa}$$

Bending stress,

$$\sigma = \frac{(32)(33.5)}{\pi(.03)^3}$$

$$= 12.6 \text{ MPa}$$

Von Mises stress

$$\sigma' = \sqrt{(12.6^2 + (3)(26)^2)}$$

$$= \mathbf{47\ MPa}$$

Based on the above results, it can be concluded that the stresses in the shaft are well below the maximum allowable stress limit.

2 STRESSES IN THE FLYWHEEL

Using references from Shigley Chapters 18 and 19, for a disc having large radius in comparison with its thickness, the stress can be assumed constant across the thickness. Stress function F is defined as follows:-

For disk of constant thickness,

$$\sigma_o = \frac{\rho r_o^2 \omega^2 (3+\nu)}{8g}$$

For disk with centre hole,

$$\sigma_r = \sigma_o \left(1 - \frac{r^2}{r_o} + \frac{r_i^2}{r_o^2} + \frac{r_i^2}{r^2} \right)$$

$$\sigma_t = r_o \left(1 - \frac{r+3\nu r^2}{3+\nu r_o^2} + \frac{r_i^2}{r_o^2} + \frac{r_i^2}{r^2} \right)$$

The maximum radial stress occurs at,

$$r = \sqrt{r_0 r_i}$$

$$\sigma_{r\,\max} = \sigma_o \left(1 - \frac{r_i}{r_o} \right)^2$$

Maximum tangential stress occurs at,

$r = r_i$

$$\sigma_{t\,\max} = \sigma_o \left(2 + \frac{2-2\nu r_o^2}{3+\nu r_o^2} \right)$$

For the 5 kW microhydro generating set, the flywheel dimensions are as follows,

Disk diameter	=	470 mm
r_o	=	235 mm
I	=	0.41 kg/m^2
Inner diameter	=	80 mm
r_i	=	40 mm

For bored disk, maximum radial stress occurred at

$$
\begin{aligned}
r &= \sqrt{(r_i\, r_o)} \\
&= \sqrt{(470)(80)} \\
&= 194 \text{ mm}
\end{aligned}
$$

Stress of flywheel without centre hole

$$\sigma_o = \frac{\rho r_o^2 \omega^2 (3+\nu)}{8g}$$

$$
\begin{aligned}
\sigma &= \frac{(7650)(.235^2)(314.2^2)(3 + .3)}{(8)(9.81)} \\
&= 17.5 \text{ MPa}
\end{aligned}
$$

Therefore, maximum radial stress

$$
\begin{aligned}
\sigma_{rmax} &= 17.5\,(1 - .04/.235)^2 \\
&= 12 \text{ MPa}
\end{aligned}
$$

Maximum tangential stress occur at $r = r_i$ (ie hub)

$$
\begin{aligned}
\sigma_{tmax} &= \frac{17.5(2 + 2\text{-}2(3)(.04)2}{3 + (.3)(.235)^2} \\
&= 17.5\,(2 + 2/3.017) \\
&= 47 \text{ MPa}
\end{aligned}
$$

Maximum Von Mises stress,

$$
\begin{aligned}
\sigma' &= \sqrt{(12^2 + 47^2 + (12)(47))} \\
&= 54 \text{ MPa}
\end{aligned}
$$

This is well within the limit of yield stress for mild steel of 250 MPa.

3 COUPLING SELECTION

The machine is design to run continuously 365 days per year. From Fenner Coupling technical data,

Power transmission		6 kW
Allowable shock factor		3
Speed		3000 RPM
Shaft size:	turbine	24 mm
	generator	22 mm

a) Service factor 1.5
(Class 1, water turbine over 16 hours of use)

b) Design power 9 kW
(6 kW x 1.5)

c) Coupling size F50 Rated power 16.7 kW

d) Bore size 11 to 32 mm

e) Normal torque 53 Nm
Maximum torque 160 Nm

Torque calculated with shock factor of 3 was 141 Nm.

f) Spacer SM16

4 PIPEWORK HEAD LOSS

From CIBSE book C, the pipe work comprises the following fittings. The highest head loss will be in the 100mm pipe, due to increased flow with small diameter pipe.

1. Butterfly valve (1 off)
2. T junction (1 off)
3. Conical strainer (1 off)
4. 180° return (1 off)
5. Reducer section and pipework friction

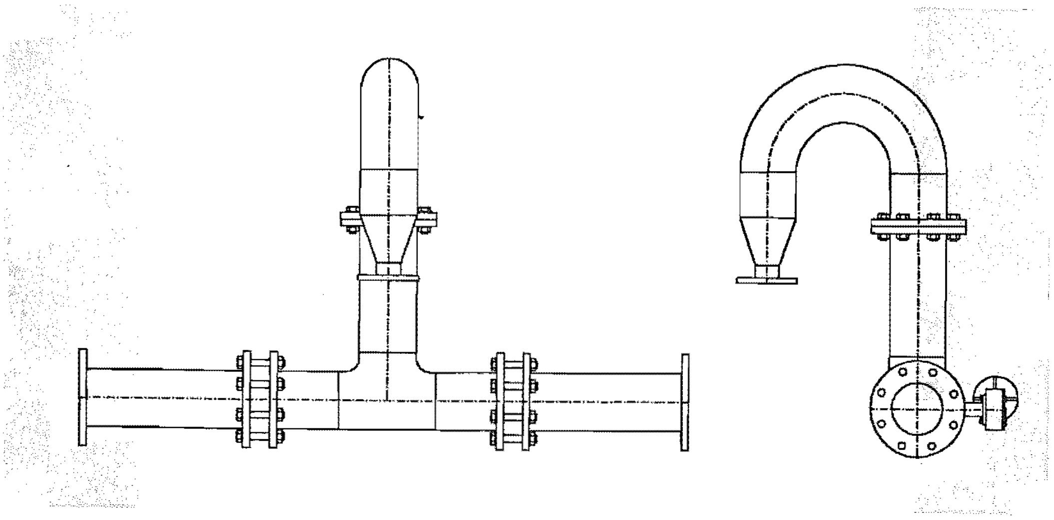


Fig A4.9 The pipework

Butterfly valve,

ξ 0.1

L_e 0.1 m

T junction,

ξ 2.0

L_e 2.2 m

Conical strainer,

ξ 2.0

L_e 2.2 m

Reducer,

ξ 0.55

L_e 0.6 m

Length of pipework,

L_e 3 m

Total equivalent length,

8 m

For galvanised metal finish, surface roughness K = 0.15 mm

K/D	=	0.0015
V	=	1.5 m/s
Re	=	1.5×10^5

From Moody Chart

f	=	0.006

Total Head loss is given by Darcy equation,

$$\Delta H = \frac{4fl}{d}\frac{v^2}{2g}$$

$$H = \frac{(4)(.006)(10)\ (1.7^2)}{(0.105)(2)(9.81)}$$

$$= 0.35 \text{ m}$$

Using the same technique, head loss in pipework for other machines are outlined in Table 8.1 of the main report.

5 DRAFT TUBE DESIGN

The function of the draft tube is to recuperate as much as possible the kinetic energy leaving the turbine. This section evaluates the optimum draft tube configuration. For the following machines, the exiting flange diameters are 50 mm, 65 mm expanding to a 100 mm pipe. All calculations are based on DS MILLER - Internal Flows [50].

The draft tube has the following configuration.

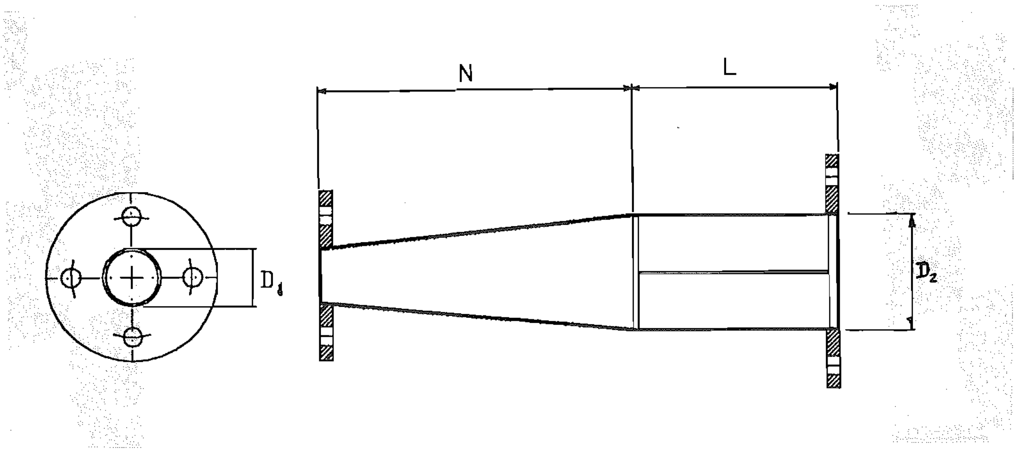


Fig A4.10 The draft tube

Sample calculations for 65x50-160,

Area ratio,

$$D_2/D_1 = 100^2/65^2 = 2.4$$

Reynolds number,

Re = 10^5 - 10^6 (This is in turbulent region)

The maximum included angle is 7° is acceptable for the range of velocities encountered. Greater included angle would cause flow separation.

For,

$$N = 300 \text{ mm}$$

$$\text{Tan}(\theta/2) = 17.5/300$$
$$\theta/2 = 3.3°$$

$$\theta = 6.6°$$

Loss coefficient for 100 mm draft tube,

$$K_d = 0.08$$

Loss coefiicient for 150mm draft tube,

$K_d \quad = \quad 0.15$

Head loss coefficient due to flow straightener is,

$K_{fs} \quad = \quad 0.1$

Head loss through draft tube is given by,

$$\textbf{\textit{Head loss}} = (K_{fs}+K_d)\frac{v^2}{2g}$$

Head loss $= (0.1 + 0.08)\ 4.4^2/(2g)$
$= 0.18$ m

Where velocity v is the draft tube inlet velocity. The results for the others machines are shown in Table 8.2 of the main report.

6 EQUIPMENT COST ESTIMATE

(As at 1991)

	COST $ (exclude GST)
-6kW Synchronous Generator	$1,200
-Brake and release mechanism	$700
-Coupling and flywheel	$700
-Turbine (KL-ISO pump)	$1,500
-Electronic Load governor	$1,300
-Baseplate	$300
-Two 150mm valves	$800
-Pipework and draft tube	$500
-Electrical protection equipment	$1,000
TOTAL	$8,000

NOTE:

The above is the cost estimate of the individual components making up the microhydro generating set. It does not include manufacturing cost. There will be other establishment incurred costs when the generating set is finally installed on site. These costs, which are dependent upon the site include:-

1. Intake
2. Penstock
3. power house and foundation
4. Electrical transmission line

APPENDIX 5

TURBINE TEST COMPUTER PROGRAM AND TEST RESULTS

COMPUTER PROGRAM FOR TURBINE TEST

```
10  OPEN "O",#1,"T50A.PRN"
20  OPEN "O",#2,"T50B.PRN"
30  '******** PROGRAM CENTRIFUGAL PUMP AS TURBINE****
40  '******** S. HENG , APRIL 1989 *************
45  '** UPDATE 18 OCTOBER BY S.HENG
46  '** PIPE LOSS DUE TO FRICTION ADDED AT INLET SIDE
50  DIM A1(30),A2(30),A3(30),A4(30),A5(30)
60  DIM B1(30),B2(30),B3(30),B4(30),B5(30),B6(30),B7(30),B8(30),B9(30),B10(30),B11(30)
70  CLS
90  INPUT "DATE:";DAT$
110 INPUT "No OF RUNS:";NRUNS
116 PRINT:INPUT"ARE THESE READINGS CORRECT? (y/n)";X$
118 IF X$="n" OR X$="N" THEN 90
130 DELZ=0
140 DI=154
150 DO=35.5
170 GOSUB 1000
175 FOR N=1 TO NRUNS
180 '*** CALCULATE EXPERIMENTAL DATA*****
185 G=9.810001
190 PI=3.14
200 DC=.214*G*.41*.75
203 L=122  'INLET FLANGE TO P TAPPING DISTANCE
207 F=.03  '** FRICTION FACTOR
210 RHO=998
215 W=RHO*G
220 RPM=3000
224 A2(N)=A2(N)*6.895
226 A3(N)=A3(N)*6.895
230 B1(N)=A4(N)/1000  '**DISCHARGE (m^3/s) ***
240 B2(N)=B1(N)^2*8/(PI^2*G)*(1/(DO/1000)^4-1/(DI/1000)^4) ' VEL HEAD***
250 B3(N)=A5(N)*DC*A1(N)*2*PI/60  '****BRAKE POWER (kW)***
255 B3(N)=B3(N)+190  '**** TIMING BELT LOSS
260 B4(N)=RPM/A1(N)  '***ACTUAL/NOM SPEED RATIO***
270 B5(N)=(A2(N)*1000-A3(N)*1000)/W+B2(N)+DELZ  '**TOT HEAD**
275 B5(N)=B5(N)-8*F*L*B1(N)^2/(DO*G*PI^2*(DO/1000)^4)  '**TOT HEAD - LOSS
280 B6(N)=B1(N)*B4(N)  '***NOM DISCHARGE***
290 B7(N)=B5(N)*B4(N)^2  '***NOM HEAD***
300 B8(N)=B3(N)*B4(N)^3  '***NOM BP****
310 B9(N)=100*B3(N)/(W*B1(N)*B5(N)) '***OVERALL EFF***
```

COMPUTER PROGRAM FOR TURBINE TEST

```
315   B10(N)=W*B1(N)*B5(N)  '***ACTUAL HYDRAULIC POWER***
330   B11(N)=W*B6(N)*B7(N)  '***NOM HYDRAULIC POWER****
485   NEXT N
490   GOTO 1490
505   PRINT:PRINT:INPUT"Would you like a PRINT-OUT (Y/N)";J$
507   IF J$="Y" OR J$="y" THEN 1980 ELSE 510
510   GOTO 6980
1000  '*********** SUBROUTINE ************
1010  '***********ROUTINE TABULATING RAW DATA ***********
1020  FOR N=1 TO NRUNS
1030  PRINT:PRINT:PRINT"RUN No.";N:PRINT
1040  INPUT"ACTUAL SPEED (RPM)";A1(N)
1050  INPUT"INLET GAUGE PRESSURE (psi)";A2(N)
1055  INPUT"SUCTION GUAGE PRESSURE (psi)";A3(N)
1060  INPUT"DISCHARGE FLOWRATE (l/s)";A4(N)
1070  INPUT"DYNAMOMETER SCALE READING";A5(N)
1080  PRINT:PRINT:INPUT "ARE THESE READINGS CORRECT (Y/N)";A$
1090  IF A$="n" OR A$="N" THEN 1030
1095  NEXT N
1110  RETURN
1490  '**********SCREEN DISPLAY***********
1495  CLS
1500  PRINT"DATE:";DAT$
1510  PRINT"CENTRIFUGAL PUMP AS TURBINE"
1520  PRINT"*********RESULTS********"
1530  PRINT "KL-ISO 50x32-200"
1540  PRINT" RUN |SPEED ||NLET |SUCTIO| FLOW | DYNO | HEAD |OUTPUT|EFFICI|HYDRAL|"
1550  PRINT" No  |ACTUAL|GUAGE |GUAGE | RATE |SCALE |  OF  |SHAFT |      |POWER |"
1560  PRINT"     |      |PRESS |PRESS |      |READIN| WATER|POWER |      |      |"
1565  PRINT"     |(RPM) |(psi) |(psi) |(l/s) |      | (m)  | (kW) | (%)  | (kW) |"
1570  PRINT"-----|------|------|------|------|------|------|------|------|------|"
1580  FOR N=1 TO NRUNS
1590  PRINT USING " ##  | #### |###.# |###.# |##.## | ##.# |###.# | #.## | ##.# |##.## |"
      ;(N),A1(N),A2(N)/6.895,A3(N)/6.895,B1(N)*1000,A5(N),B5(N),B3(N)/1000,B9(N),B10(N)/1000
1600  PRINT USING "     | #### |      |      |##.## |      |###.# | #.## | ##.# |##.## |";RPM,B6(N)*1000,B7(N),B8(N)/1000,B9(N),B11(N)/1000
1610  NEXT N
1620  GOTO 505
1980  '******PRINT-OUT*******
1985  WIDTH "LPT1:",120
2100  LPRINT CHR$(14);TAB(7) "CENTRIFUGAL PUMP AS TURBINE"
```

COMPUTER PROGRAM FOR TURBINE TEST

```
2200 LPRINT TAB(27) "********RESULTS********":LPRINT
2250 LPRINT:LPRINT"        DATE:";DAT$
2300 LPRINT "        PUMP SIZE: KL-ISO 50x32-200"
2310 LPRINT "        MODEL No: AKP00306"
2320 LPRINT "        SERIAL No: 0164C  "
2350 LPRINT "        RPM=3000":LPRINT
2400 LPRINT CHR$(15)
2500 LPRINT"          RUN  | DISCHARGE |  HEAD   |  SHAFT  | EFFICIENCY| HYDRAULIC |"
2530 LPRINT"          No   | FLOWRATE  | OF WATER |  POWER  |           |  POWER   |"
2550 LPRINT"               |   (l/s)   |   (m)   |   (kW)  |    (%)    |   (kW)   |"
2900 LPRINT"          -------|-----------|-----------|-----------|-----------|-----------|"
3000 FOR N=1 TO NRUNS
3100 LPRINT USING "           ##  |  ##.#   |  ###.#  |  ##.##  |   ##.#    |  ##.##   |";(N),B6(N)*1000,B7(N),B8(N)/1000,B9(N),B11(N)/1000
3150 LPRINT"               |         |         |         |           |          |"
3300 NEXT N
3400 PRINT:PRINT:INPUT"Would you like another copy (y/n)";C$
3420 IF C$="N" OR C$="n" OR C$="0" THEN 4900
3440 PRINT:PRINT:INPUT"Adjust paper then PRESS ANY KEY TO PRINT";W$
3460 IF W$="Q" THEN 4000 ELSE 1980
4000 '******RAW DATA PRINT-OUT******
4005 WIDTH "LPT1:",120
4010 LPRINT CHR$(14);TAB(7)"CENTRIFUGAL PUMP AS TURBINE"
4020 LPRINT TAB(27)"****** RAW DATA ******"
4030 LPRINT:LPRINT"        DATE:";DAT$
4040 LPRINT"        PUMP SIZE: KL ISO 50x32-200"
4043 LPRINT"        MODEL No: AKP00306"
4047 LPRINT"        SERIAL No: 0164C  ":LPRINT
4050 LPRINT CHR$(15)
4100 LPRINT"           RUN  |  ACTUAL  | INL GUAGE | SUC GUAGE | DISCHARGE | DYN SCALE |"
4200 LPRINT"           No   |  SPEED   | PRESSURE  | PRESSURE  | FLOWRATE  |  READING  |"
4300 LPRINT"                |  (RPM)   |   (psi)   |   (psi)   |   (l/s)   |           |"
4400 LPRINT"           ------|-----------|-----------|-----------|-----------|-----------|"
4500 FOR N=1 TO NRUNS
4600 LPRINT USING"           ##  |  ####   |  ###.#  |  ###.#  |   ##.#    |  ##.#    |";(N),A1(N),A2(N)/6.895,A3(N)/6.895,B1(N)*1000,A5(N)
4650 LPRINT"                |         |         |         |           |          |"
4800 NEXT N
4900 PRINT:PRINT:INPUT"Would you like a copy of the RAW DATA";E$
5000 IF E$="N" OR E$="n" THEN 6000
5100 PRINT:PRINT:INPUT"Adjust paper then ENTER";F$
5200 IF F$="Q" THEN 6000 ELSE 4000
```

COMPUTER PROGRAM FOR TURBINE TEST

```
6980 '******PRINTING TO FILE********
6985 WIDTH #1,120
7100 PRINT #1,CHR$(14);TAB(7) "CENTRIFUGAL PUMP AS TURBINE"
7200 PRINT #1,TAB(27) "********RESULTS********":PRINT #1,
7250 PRINT #1,:PRINT #1,"        DATE:";DAT$
7300 PRINT #1,"       PUMP SIZE: KL-ISO 50x32-200"
7310 PRINT #1,"       MODEL No: AKP00306"
7320 PRINT #1,"       SERIAL No: 0164C  "
7350 PRINT #1,"       RPM=3000":PRINT #1,
7400 PRINT #1,CHR$(15)
7500 PRINT #1,"          RUN  I DISCHARGE I  HEAD   I  SHAFT  I EFFICIENCYI HYDRAULIC I"
7530 PRINT #1,"          No   I FLOWRATE  I OF WATER I  POWER  I           I  POWER  I"
7550 PRINT #1,"               I  (l/s)  I   (m)   I   (kW)  I   (%)   I   (kW)  I"
7900 PRINT #1,"          -------I-----------I-----------I-----------I-----------I-----------I"
8000 FOR N=1 TO NRUNS
8100 PRINT #1,USING "          ##  I  ##.#   I  ###.#  I  ##.##   I  ##.#   I  ##.##  I";(N),B6(N)*1000,B7(N),B8(N)/1000,B9(N),B11(N)/1000
8150 PRINT #1,"               I         I         I         I         I         I"
8300 NEXT N
9000 '********* PRINTING RAW DATA TO FILE*****
9005 WIDTH #2,120
9010 PRINT #2,CHR$(14);TAB(17)"CENTRIFUGAL PUMP AS TURBINE"
9020 PRINT #2,"               ****** RAW DATA *******"
9030 PRINT #2,: PRINT #2,"        DATE:";DAT$
9040 PRINT #2,"       PUMP SIZE: KL ISO 50x32-200"
9043 PRINT #2,"       MODEL No: AKP00306"
9047 PRINT #2,"       SERIAL No: 0164C  ":PRINT #2,
9050 PRINT #2,CHR$(15)
9100 PRINT #2,"           RUN  I  ACTUAL   I INL GUAGE I SUC GUAGE I DISCHARGE I DYN SCALE I"
9200 PRINT #2,"           No   I  SPEED    I PRESSURE  I PRESSURE  I FLOWRATE  I  READING  I"
9300 PRINT #2,"                I  (RPM)  I  (psi)  I  (psi)  I  (l/s)  I         I"
9400 PRINT #2,"           ------I-----------I-----------I-----------I-----------I-----------I"
9500 FOR N=1 TO NRUNS
9600 PRINT #2,USING"    ##  I  ####   I  ###.#  I  ###.#   I  ##.#   I  ##.#   I";(N),A1(N),A2(N)/6.895,A3(N)/6.895,B1(N)*1000,A5(N)
9650 PRINT #2,"                I         I         I         I         I         I"
9800 NEXT N
```

TURBINE TEST DATA

TURBINE TEST (T50ASM)	RUN	DISCHARGE	HEAD	SHAFT	EFFICIENCY	HYDRAULIC
DATE:19 OCT 90	No		OF WATER	POWER		POWER
PUMP SIZE: KL-ISO 50x32-200		(l/s)	(m)	(kW)	(%)	(kW)
MODEL No: AKP00306						
SERIAL No: 0164C	1	10.2	181	7.84	43.3	18.1
RPM=3000	2	9.5	163.4	6.56	43.2	15.19
	3	8.8	146	5.41	43.1	12.54
	4	8.1	130	4.25	41.5	10.25
	5	7.3	116.1	3.13	37.5	8.33
	6	6.6	105.3	2.09	30.6	6.81
	7	5.9	95.3	1.35	24.6	5.49

TURBINE TEST (T65X)	RUN	DISCHARGE	HEAD	SHAFT	EFFICIENCY	HYDRAULIC
DATE:20 FEB 90	No		OF WATER	POWER		POWER
PUMP SIZE: KL-ISO 65x40-200		(l/s)	(m)	(kW)	(%)	(kW)
MODEL No: AKP00706C						
SERIAL No: 01576	1	8.5	65.5	1.07	19.6	5.47
RPM=3000	2	10	70.8	2.58	37.2	6.94
	3	11.3	76.4	4.06	48	8.46
	4	12.8	83.1	5.75	55.4	10.39
	5	14.1	90.5	7.4	59.4	12.46
	6	15.2	99.1	9.17	62.1	14.77
	7	16.7	110.8	11.61	63.9	18.15
	8	18.5	123.2	14.04	63	22.28
	9	19.6	135.4	16.46	63.3	26.01
	10	20.4	142.8	17.96	63.1	28.45

TURBINE TEST (T65X185)	RUN	DISCHARGE	HEAD	SHAFT	EFFICIENCY	HYDRAULIC
DATE:20 FEB 90	No		OF WATER	POWER		POWER
PUMP SIZE: KL-ISO 65x40-200[185]		(l/s)	(m)	(kW)	(%)	(kW)
MODEL No: AKP00706C						
SERIAL No: 01576	1	6.8	43.9	0.5	17.1	2.91
RPM=3000	2	8	47.9	1.32	35	3.78
	3	9.3	53.2	2.2	45.6	4.83
	4	10.6	60	3.16	50.6	6.25
	5	12.1	68.7	4.46	54.7	8.17
	6	12.6	73.1	4.96	55.1	9.01
	7	14.2	83.7	6.45	55.4	11.63
	8	15.2	92.5	7.67	55.7	13.77
	9	16.1	100.1	8.64	54.8	15.78

TURBINE TEST DATA

TURBINE TEST (T50ASM)	RUN	DISCHARGE	HEAD	SHAFT	EFFICIENCY	HYDRAULIC
DATE:19 OCT 90	No		OF WATER	POWER		POWER
PUMP SIZE: KL-ISO 50x32-200		(l/s)	(m)	(kW)	(%)	(kW)
MODEL No: AKP00306						
SERIAL No: 0164C	1	10.2	181	7.84	43.3	18.1
RPM=3000	2	9.5	163.4	6.56	43.2	15.19
	3	8.8	146	5.41	43.1	12.54
	4	8.1	130	4.25	41.5	10.25
	5	7.3	116.1	3.13	37.5	8.33
	6	6.6	105.3	2.09	30.6	6.81
	7	5.9	95.3	1.35	24.6	5.49

TURBINE TEST (T65X)	RUN	DISCHARGE	HEAD	SHAFT	EFFICIENCY	HYDRAULIC
DATE:20 FEB 90	No		OF WATER	POWER		POWER
PUMP SIZE: KL-ISO 65x40-200		(l/s)	(m)	(kW)	(%)	(kW)
MODEL No: AKP00706C						
SERIAL No: 01576	1	8.5	65.5	1.07	19.6	5.47
RPM=3000	2	10	70.8	2.58	37.2	6.94
	3	11.3	76.4	4.06	48	8.46
	4	12.8	83.1	5.75	55.4	10.39
	5	14.1	90.5	7.4	59.4	12.46
	6	15.2	99.1	9.17	62.1	14.77
	7	16.7	110.8	11.61	63.9	18.15
	8	18.5	123.2	14.04	63	22.28
	9	19.6	135.4	16.46	63.3	26.01
	10	20.4	142.8	17.96	63.1	28.45

TURBINE TEST (T65X185)	RUN	DISCHARGE	HEAD	SHAFT	EFFICIENCY	HYDRAULIC
DATE:20 FEB 90	No		OF WATER	POWER		POWER
PUMP SIZE: KL-ISO 65x40-200[185]		(l/s)	(m)	(kW)	(%)	(kW)
MODEL No: AKP00706C						
SERIAL No: 01576	1	6.8	43.9	0.5	17.1	2.91
RPM=3000	2	8	47.9	1.32	35	3.78
	3	9.3	53.2	2.2	45.6	4.83
	4	10.6	60	3.16	50.6	6.25
	5	12.1	68.7	4.46	54.7	8.17
	6	12.6	73.1	4.96	55.1	9.01
	7	14.2	83.7	6.45	55.4	11.63
	8	15.2	92.5	7.67	55.7	13.77
	9	16.1	100.1	8.64	54.8	15.78

TURBINE TEST DATA

TURBINE TEST (T65HASM)	RUN No	DISCHARGE (l/s)	HEAD OF WATER (m)	SHAFT POWER (kW)	EFFICIENCY (%)	HYDRAULIC POWER (kW)
DATE:19 OCT 89						
PUMP SIZE: KL-ISO 65x50-160						
MODEL No: AK0606/C						
SERIAL No: 039-5217	1	15.9	75.8	8.1	68.4	11.83
RPM=3000	2	15	69.5	6.95	68.3	10.17
	3	14	64	6.13	70	8.76
	4	13	58.8	5.2	69.5	7.48
	5	12	54.1	4.48	70.4	6.36
	6	11	50.2	3.65	67.4	5.42
	7	10	46.6	2.89	63.2	4.57
	8	9	43.1	2.28	59.9	3.81
	9	8	40.3	1.57	49.5	3.18
	10	7	37.7	0.94	36	2.6

TURBINE TEST (T80XF)	RUN No	DISCHARGE (l/s)	HEAD OF WATER (m)	SHAFT POWER (kW)	EFFICIENCY (%)	HYDRAULIC POWER (kW)
DATE:20 FEB 90						
PUMP SIZE: KL-ISO 80x65-160						
MODEL No: AKP01106C						
SERIAL No: AK01714	1	11.7	32	0.57	15.6	3.67
RPM=3000	2	14	33.9	1.71	37	4.63
	3	16.1	35.7	2.79	49.4	5.65
	4	19	39.3	4.32	59.3	7.29
	5	21.7	43.7	6.46	69.5	9.29
	6	24.2	49.1	8.6	73.9	11.63
	7	25.5	52.4	9.81	75	13.08
	8	25.9	53	9.94	76	13.42
	9	26.9	55.7	11.09	75.7	14.65
	10	27.9	57.7	11.88	75.5	15.73
	11	28.6	59.8	12.73	75.9	16.77
	12	29.2	61.6	13.43	76.1	17.64
	13	29.8	64.2	14.41	77	18.72
	14	30.8	67	15.53	77	20.17
	15	31.8	69.7	16.69	77.1	21.66
	16	32.8	73.4	17.96	76.2	23.57
	17	33.3	75.9	18.85	76.2	24.74

TURBINE TEST DATA

TURBINE TEST (T80X142)
DATE:20 FEB 90
PUMP SIZE: KL-ISO 80x65-160[142]
MODEL No: AKP01106C
SERIAL No: AK01714
RPM=3000

RUN No	DISCHARGE (l/s)	HEAD OF WATER (m)	SHAFT POWER (kW)	EFFICIENCY (%)	HYDRAULIC POWER (kW)
1	11.1	19.9	0.69	31.5	2.17
2	12.4	21.7	1.16	44.1	2.64
3	13.5	23.5	1.58	51.1	3.1
4	14.5	25.7	2.1	57.9	3.64
5	15.7	27.5	2.56	60.7	4.21
6	17	31.1	3.35	64.6	5.18
7	18.6	35.3	4.4	68.6	6.41
8	19.8	38.9	5.32	70.5	7.55
9	21.2	43.4	6.31	70	9.02
10	22.2	46.2	6.99	69.6	10.04
11	23.2	49.6	7.7	68.4	11.27
12	23.8	51.7	8.24	68.5	12.03
13	24.8	55.2	9.08	67.7	13.41

TURBINE TEST (T80ASM)
DATE:19 OCT 89
PUMP SIZE: KL-ISO 80x65-125
MODEL No: AK1006C
SERIAL No: 039-8228
RPM=3000

RUN No	DISCHARGE (l/s)	HEAD OF WATER (m)	SHAFT POWER (kW)	EFFICIENCY (%)	HYDRAULIC POWER (kW)
1	30.2	31.8	7.39	78.5	9.42
2	29.4	30.3	6.86	78.7	8.71
3	28.5	28.7	6.36	79.3	8.02
4	27.7	27.3	5.83	78.7	7.4
5	26.8	26.1	5.32	77.8	6.83
6	25.9	24.8	4.81	76.3	6.3
7	25	23.7	4.38	75.5	5.8
8	24.1	22.4	3.88	73.3	5.29
9	23.2	21.4	3.49	71.8	4.86
10	22.3	20.5	3.04	68.1	4.46
11	21.3	19.6	2.64	64.6	4.08
12	20.4	18.7	2.3	61.7	3.73
13	19.4	18	1.94	56.7	3.41
14	18.4	17.3	1.56	50.2	3.12
15	17.4	16.5	1.17	41.4	2.82
16	16.5	15.7	0.85	33.5	2.53
17	15.5	14.9	0.53	23.5	2.26